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PRACTICAL INSTRUCTIONS
FOR USING THE
STEAM ENGINE INDICATOR

CROSBY STEAM GAGE & VALVE CO.
BOSTON, MASS., U. S. A.

JUN 6 1910

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PRACTICAL INSTRUCTIONS

RELATING TO THE CONSTRUCTION AND USE OF THE

STEAM ENGINE INDICATOR

PART I

GENERAL DESIGN AND CONSTRUCTION OF STEAM ENGINE INDICATORS

SPECIAL DESIGN, CONSTRUCTION AND USE
OF THE

CROSBY INDICATOR

PART II

APPLICATIONS OF THE INDICATOR

PART III

PROPERTIES OF STEAM AND OF PERFECT GASES

DESIGN OF A PLAIN SLIDE VALVE

VALVE SETTING ON PLAIN SLIDE VALVE ENGINES AND ON CORLISS ENGINES

PARTS II AND III
BY EDWARD F. MILLER
Professor of Steam Engineering
MASSACHUSETTS INSTITUTE OF TECHNOLOGY

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TILDEN FOUNDATIONS.
1910
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Crosby Steam Gage and Valve Company

The Barta Press, Boston

INTRODUCTION

The purpose of this book is to enable any engineer, First, To understand the design, construction and use of the Crosby Steam Engine Indicator. Second, To make suitable preparation for applying it to a steam engine, and attaching the mechanism for operating the paper drum. Third, To take diagrams, read them intelligently, and, after some experience, to deduce from them such information concerning the working of an engine as a good instrument skilfully applied and handled is capable of revealing to the studious and observing mind.

Crosby Steam Gage and Valve Company

Boston, Jan. 1, 1910

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PRACTICAL INSTRUCTIONS

RELATING TO THE CONSTRUCTION
AND USE OF THE

STEAM ENGINE INDICATOR

PART I

CHAPTER I

THE STEAM ENGINE INDICATOR

The steam engine indicator, invented by James Watt, and long kept secret, was for many years after its secret became known, strangely neglected by most makers and users of steam engines.

The earlier forms of the instrument, which preceded that invented by Richards, were so imperfect and so ill-adapted to engines running at other than very low speeds, that their indications were often misleading, more often unintelligible, and seldom of much value beyond revealing the point of stroke at which the valves opened and closed: a most valuable service, alone worth the cost of an indicator, but only a small part of the service to be obtained from a really good instrument.

The general principles on which the best type of the steam engine indicator is designed, may be briefly stated as follows:

A piston of carefully determined area is nicely fitted into a cylinder so that it will move up and down without sensib friction. The cylinder is open at the bottom and fitted so that it may be attached to the cylinder of a steam engine and have free communication with its interior, by which arrangement the under side of the piston is subjected to the varying pressure of the steam acting therein. The upward movement of the piston — due to the pressure of the steam — is resisted by a spiral spring of known resilience. A piston rod projects upward through the cylinder cap and moves a lever having at its free end a pencil point, whose vertical movement bears a constant ratio to that of the piston. A drum of cylindrical form and covered with paper is attached to the cylinder in such a manner that the pencil point may be brought in contact with its surface, and thus record any movement of either paper or pencil; the drum is given a horizontal motion coincident with and bearing a constant ratio to the movement of the piston of the engine. It is moved in one direction by means of a cord attached to the crosshead and in the opposite direction by a spring within itself.

When this mechanism is properly adjusted and free communication is opened with the cylinder of a steam engine in motion, it is evident that the pencil will be moved vertically by the varying pressure of steam under the piston, and as the drum is rotated by the reciprocating motion of the engine, if the pencil is held in contact with the moving paper during one revolution of the engine, a figure or diagram will be traced representing the pressure of steam in the cylinder; the upper line showing the pressure urging the engine piston forward, and the lower the pressure retarding its movement on the return stroke.

To enable the engineer to more correctly interpret the nature of the pressures, the line showing the atmospheric pressure is drawn in its relative position, which indicates whether the pressure at any part is greater or less than that *of the atmosphere.*

From such a diagram may be deduced many particulars which are of supreme importance to engine builders, engineers, and the owners of steam plants.

WHAT IS THE GOOD OF AN INDICATOR?

This question was asked by a young engineer who had come to examine and purchase a Crosby indicator, with a view to rendering his services of greater value to his employer, by a knowledge and use of that instrument. His question was overheard by the proprietor of a large establishment in the city of Worcester, Mass., who took occasion to reply as follows:

"I will tell you what good an indicator did at our works. Our steam engine was not giving sufficient power for our business, and we expected to be obliged to procure a larger one. A neighbor suggested that we have our engine indicated to see if we were getting the best service obtainable from it. This was done with a Crosby indicator, and the result was that when the valves were properly adjusted and other slight changes made we had *ample power*, and the improved condition of the engine made a reduction in our coal bills during the following year of $500."

Another case: An expert engineer was called to indicate several locomotives just completed by one of our prominent locomotive builders, who had in use a large Corliss engine, which had been running only a few months. When the locomotives were indicated, the proprietor proposed that the indicator be applied to the Corliss engine, the engineer of which remarked, "Guess you'll find her all right, as she's running fine."

The first card showed that *nearly all the work was being done at one end of the cylinder.* The valves were changed and a great improvement was apparent in the running of the engine, while the actual consumption of coal was reduced from an average of 3,370 pounds per day, before the change was made, to 2,338 pounds afterwards.

These two instances are valuable in showing "the good of an indicator."

Items of Information to be obtained by the use of the Indicator

The arrangement of the valves for admission, cut-off, release and compression of steam.

The adequacy of the ports and passages for admission and exhaust, and when applied to the steam chest, the adequacy of the steam pipes.

The suitableness of the valve motion in point of rapidity at the right time.

The quantity of power developed in the cylinder, and the quantity lost in various ways: by wire drawing, by back pressure, by premature release, by mal-adjustment of valves, by leakage, etc.

It is useful to the designers of steam engines, as by properly combining the cards the effective pressure on the piston at any point can be determined, and from this the rotative effect or the rotating force acting at right angles to the crank can be calculated.

Taken in combination with measurements of feed water and the condensation and measurement of the exhaust steam, with the amount of fuel used, the indicator furnishes many other items of importance when the economical generation and use of steam are considered.

For every one of these purposes it is important that the diagrams traced by the indicator should truly represent the path of the piston and the pressure exerted on both sides of the piston at every point of that path.

INDICATOR DIAGRAMS

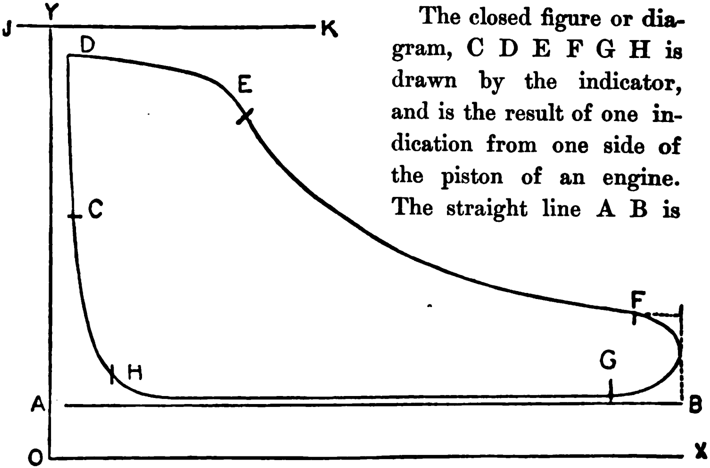
The degree of excellence to which steam engines of the present time have been brought is due more to the use of the indicator than to any other cause, as a careful study of *indicator diagrams* taken under different conditions of load, pressure, etc., is the only means of becoming familiar with the action of steam in an engine, and of gaining a definite knowledge of the various changes of pressure that take place in the cylinder.

An indicator diagram is the result of two movements, namely: a horizontal movement of the paper in exact correspondence with the movement of the piston, and a vertical movement of the pencil in exact ratio to the pressure exerted in the cylinder of the engine; consequently, it represents by its length the stroke of the engine on a reduced scale, and by its height at any point, the pressure on the piston at the corresponding point in the stroke. The shape of the diagram depends altogether upon the manner in which the steam is admitted to and released from the cylinder of the engine; the variety of shapes given from different engines, and by the same engine under different circumstances, is almost endless, and it is in the intelligent and careful measurement of these that the true value of the indicator is found, and no one at the present day can claim to be a competent engineer who has not become familiar with the use of the indicator, and skilful in turning to practical advantage the varied information which it furnishes.

A diagram shows the pressure acting on one side of the piston only, during both the forward and return stroke, whereon all the changes of pressure may be properly located, studied, and measured. To show the corresponding pressures on the other side of the piston, another diagram must be taken from the other end of the cylinder. When the three-way cock is used, the diagrams from both ends are usually taken on the same paper, as in Fig. 2.

ANALYSIS OF THE DIAGRAM

The names by which the various points and lines of an indicator diagram are known and designated are given on *the page following.* See Fig. 1.



The closed figure or diagram, C D E F G H is drawn by the indicator, and is the result of one indication from one side of the piston of an engine. The straight line A B is also drawn by the indicator, but at a time when steam connection with the engine is closed, and both sides of the indicator piston are subjected to atmospheric pressure only.

FIG. 1

The straight lines O X, O Y, and J K, when required, are drawn by hand as explained below, and may be called reference lines.

DIAGRAM LINES EXPLAINED

The admission line C D shows the rise in pressure due to the admission of steam to the cylinder by the opening of the steam valve. If the steam is admitted quickly when the engine is about on the dead-center this line will be nearly vertical.

The steam line D E is drawn when the steam valve is open and steam is being admitted to the cylinder.

The point of cut-off E is the point where the admission of steam is stopped by the closing of the valve. It is sometimes difficult to determine the exact point at which the cut-off takes place. It is usually located where the outline of *the diagram changes* its curvature from convex to concave.

The expansion curve E F shows the fall in pressure as the steam in the cylinder expands behind the moving piston of the engine.

The point of release F shows when the exhaust valve opens.

The exhaust line F G represents the loss in pressure which takes place when the exhaust valve opens at or near the end of the stroke.

The back pressure line G H shows the pressure against which the piston acts during its return stroke. On diagrams taken from non-condensing engines it is either coincident with or above the atmospheric line, as in Fig. 1. On cards taken from a condensing engine, however, it is found below the atmospheric line, and at a distance greater or less, according to the vacuum obtained in the cylinder.

The point of exhaust closure H is the point where the exhaust valve closes. It cannot be located very definitely, as the change in pressure is at first due to the gradual closing of the valve.

The compression curve H C shows the rise in pressure due to the compression of the steam remaining in the cylinder after the exhaust valve has closed.

The atmospheric line A B is a line drawn by the pencil of the indicator when its connections with the engine are closed and both sides of the piston are open to the atmosphere. This line represents on the diagram the pressure of the atmosphere, or zero of the steam gage.

REFERENCE LINES EXPLAINED

The zero line of pressure, or line of absolute vacuum, O X, is a reference line, and is drawn by hand, $14\frac{7}{10}$ pounds by the scale, below and parallel with the atmospheric line. It represents a perfect vacuum, or absence of all pressure.

The line of boiler pressure J K is drawn by hand parallel to the atmospheric line and at a distance from it by the scale of an ordinary horse, and is now simply an arbitrary standard. It is equal to 33,000 foot-pounds exerted during one minute of time, or 550 foot-pounds during one second.

Indicated horsepower is the horsepower of an engine as found by the use of a steam engine indicator, and is thus expressed: I. H. P.

Net horsepower is the indicated horsepower of an engine, less the horsepower which is consumed in overcoming its own friction.

Wire drawing, as applied to steam, is the reducing of its pressure, due to its flowing through restricted or crooked pipes and passages.

Absolute pressure is pressure reckoned from absolute vacuum; in other words, it is the pressure of any fluid as shown by a pressure gage, with the weight or pressure of the atmosphere added thereto.

Initial forward pressure in a cylinder is the pressure acting on the piston at or near the beginning of the forward stroke.

Terminal forward pressure is the pressure above the line of perfect vacuum that would exist at the end of the stroke if the steam had not been released earlier. It may be found by continuing the expansion curve to the end of the diagram, as in Fig. 1 at F, or it may be taken at the point of release. This pressure is always measured from the line of perfect vacuum, hence it is the absolute terminal pressure.

Mean effective pressure (written as M. E. P.) is that equivalent constant pressure which will do the same amount of work on the piston per stroke as is done by the varying pressure shown by the indicator card. This may be calculated by dividing the area of the card by the length and multiplying by the scale of the spring used in the indicator, or it may be obtained by taking the average of a number of pressures measured between the top and the bottom lines of the card at equidistant intervals along the length of the card. This is illustrated more fully on page 85.

Piston displacement is the space in the cylinder swept through by the piston in its travel. It is reckoned in cubic feet, and is found by multiplying the net area of the piston in square feet by the length of stroke in feet, allowance being made for the piston rod.

Clearance is all the waste room or space at either end of the cylinder, between its head and the piston when on a dead center, including the counterbore and the ports, up to the face of the closed valves. Clearance is generally given as a percentage of the piston displacement. Many of the modern steam engines have clearances no greater than 2¼ per cent.

The number of expansions is the number of times the steam admitted up to cut-off in the first cylinder of a multiple expansion engine has increased in volume up to release in the last cylinder.

If the release be considered to come at the end of the stroke the number of expansions for any engine may be calculated from the diameter of the cylinders and the per cent of cut-off in the first cylinder.

Thus — cylinders of a triple engine are 9″ — 16″ — 24″ × 30″ stroke; cut-off at ⅓ stroke in high pressure cylinder.

As the strokes are the same on all the cylinders of this engine the volumes of the cylinders will be as the squares of their diameters.

If the high pressure cylinder took steam to the end of its stroke the number of expansions would be $\frac{24^2}{9^2}$, but the steam expands three times in the high, so the total expansion in the engine is $\frac{24^2}{9^2} \times 3 = 21.3$.

Sensible heat is the temperature of any body, as air, water, or steam, which may be measured by the thermometer.

Specific heat is the quantity of heat required to raise o unit of weight of the substance through one degree of te perature, measured in thermal units.

The unit of heat, or thermal unit, is the quantity of heat required to raise the temperature of one pound of water from 62° to 63° F. This is often called the British Thermal Unit (written B. t. u.), to distinguish it from the German and the French standard, which is larger.

Mechanical equivalent of heat. It has been found by experiment that if one pound of pure water at 62° F. be raised to 63° F., energy is exerted equivalent to lifting seven hundred and seventy-eight (778) pounds one foot high, or one pound seven hundred and seventy-eight (778) feet high. This energy is called the *mechanical equivalent of one thermal unit of heat,* and it is usually designated by the letter J, and its reciprocal or $\frac{1}{778}$ by A.

Saturated steam. When steam is formed in a closed vessel in contact with its own liquid, it is said to be saturated and it will have a certain definite pressure and density corresponding to each different temperature. If, at the same time, the steam contains no liquid in suspension, it is said to be *dry* and *saturated.*

Superheated steam. If, after *all* the liquid has been converted into steam, more heat be added, the temperature will rise and the steam is said to be superheated, because its temperature will be greater than that corresponding to *saturated* steam of the *same pressure.* This difference is the number of degrees of superheat.

Priming. It is possible for saturated steam to hold particles of water in suspension. The steam is then said to be primed or wet. The amount of water so suspended may vary from zero to 5 per cent. Almost all boilers in which the steam is not in contact with the hot gases give steam primed from 0.1 per cent to 3 per cent.

Thermal efficiency of an engine. An engine having a thermal efficiency of 100 per cent would require $\frac{33000}{778} =$ *42.42 B. t. u. per H. P.* per minute. An engine which used

250 B. t. u. per H. P. per minute thus has a thermal efficiency of $\frac{42.42}{250} = 0.169$, or 16.9 per cent.

Mechanical efficiency of an engine. This is the ratio of the power delivered at the fly-wheel or shaft to that calculated from the indicator cards. This ratio may be about 90 per cent.

Temperature. There are a number of different thermometric scales. The Fahrenheit and the Centigrade are the two most generally used. In the Fahrenheit system melting ice is taken as 32°, and boiling water, or steam at 14.7 pounds pressure, is taken as 212° F. In the Centigrade system melting ice is taken as 0° and steam at 14.7 pounds pressure as 100° C. A Centigrade reading may be converted into the Fahrenheit scale by multiplying the Centigrade reading by $\frac{9}{5}$ and then adding 32. The zero of each of these two systems is purely arbitrary.

It is possible to construct a thermodynamic scale of temperature based on thermodynamic laws. The zero of this scale, called the *absolute zero*, is 459.5° below the zero of the Fahrenheit scale and 273.1° below the zero of the Centigrade scale. The zero obtained by what is known as the air thermometer agrees very closely with the zero of the thermodynamic scale. The absolute temperature corresponding to 100° F. is $100 + 459.5 = 559.5°$.

Isothermal expansion. If, while a substance expands, an amount of heat sufficient to keep the temperature constant be added, the expansion is said to be isothermal. The pressure may or may not change during the expansion. In the case of a mixture of water and steam the pressure would remain constant as long as any water was present. In the case of a perfect gas the pressure would decrease in the same proportion as the volume increased.

Isothermal compression. This is the reverse of the expansion, and heat has to be abstracted during compression in order to keep the temperature constant. An amount of work has to be done on the vapor or substance in compressing it equal to that which would be obtained as useful work during an isothermal expansion between the same limits.

Adiabatic expansion. A reversible adiabatic expansion is such an expansion as would take place between cut-off and release in a steam engine or compressed air engine, if there was no heat given to or taken from the metal forming the cylinder and the piston. It is an expansion during which no heat is either added or taken away as heat. As work is done on the piston it must be paid for in some way. This work comes from the internal energy the substance had at the beginning of the expansion, and if during the expansion 7,780 foot-pounds of external work or useful work were done on the piston, then the internal energy at the end of this expansion would be 7,780 foot-pounds less than at the beginning.

During an adiabatic expansion both the pressure and the temperature drop as the volume increases. If the substance, when compressed adiabatically, goes back over the same path along which it expanded, the expansion or compression takes place at constant entropy.

Entropy is the name given to a term representing the value of a ratio, or the value of the summation of a number of such ratios. This term appears frequently in the calculation of engineering problems. An example will illustrate. Suppose a gas to expand at a constant temperature of 100° F., and that during this expansion 5 B. t. u. are added. The *increase* in entropy is $\frac{5}{100 + 459.5} = 0.0089$, and is found by dividing the heat added by the constant absolute temperature at which it was added.

Suppose that instead of the *temperature* remaining constant, the *temperature* increased 10° while the 5 B. t. u. were *being added.* An approximation to the increase in entropy could be made as follows: Each heat unit caused an increase in temperature of 2°, so that the average absolute temperature, while the first heat unit was being added, was $101 + 459.5 = 560.5°$. The increase in entropy due to this one heat unit is approximately $\frac{1}{560.5}$; the increase due to the addition of the second heat unit at an average absolute temperature of 562.5 is $\frac{1}{562.5}$, and the total increase is approximately equal to the sum of $\frac{1}{560.5} + \frac{1}{562.5} + \frac{1}{564.5} + \frac{1}{566.5} + \frac{1}{568.5}$. This amount is smaller than in the case where the temperature remained constant. To get the correct value of the increase in entropy for this case, we should have a summation of a great many terms, adding, instead of one heat unit, an infinitesimal amount of heat each time, and then dividing each infinitesimal amount of heat by the absolute temperature at which it was added.

It is evident that entropy is always figured as the increase or the decrease between two conditions. In other words there is no zero entropy.

CHAPTER II

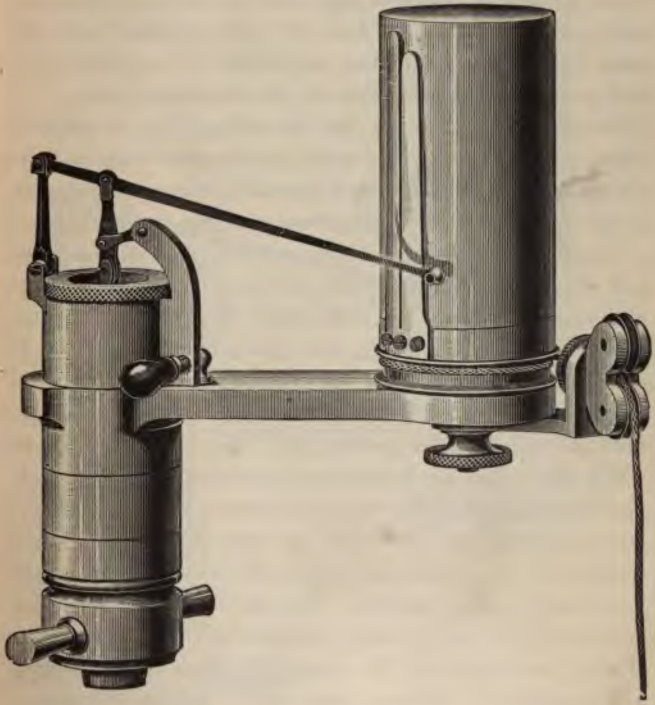
THE CROSBY STEAM ENGINE INDICATOR

The Crosby Steam Engine Indicator is designed and constructed to meet the exacting requirements of modern steam engineering. During the last few years, under the keen search and exhaustive tests of eminent engineers, the practice in this department of science has undergone important changes, tending to establish more correct methods and thereby to reach more accurate results; especially is this true in the use and scope of the indicator, so that the work done with such instruments in former times seems coarse and crude when compared with the more exact attainment of the present.

Educators in the scientific schools of both Europe and America have seen the importance of more exact knowledge and instruction in the technical sciences; and the great achievements of recent years in the construction of buildings, ships, armaments and machines attest the thoroughness with which research in these departments has been prosecuted; in none has there been greater progress made than in mechanical and steam engineering.

A knowledge of these facts has kept us on the alert in the manufacture of all our steam appliances, and especially in that of the steam engine indicator. Within a recent time we have made important improvements, which, as we believe, place it far in advance of any other instrument of its kind. Radical changes in design, more perfect mechanical construction, due to the use of improved and specialized machinery, and careful selection of metals for the different parts, have all contributed to this favorable *result.*

The movements of piston and pencil point are perfectly parallel, the movement of the pencil point is also exactly parallel with the axis of the drum. This accuracy is secured by mathematically correct design and careful workmanship.



The rating of the springs by our newly constructed testing apparatus, which embodies all the valuable aids to exactness which have yet been discovered, is nearer perfection than could have been attained, or even expected, until within *a very recent* time.

CROSBY STANDARD STEAM ENGINE INDICATOR

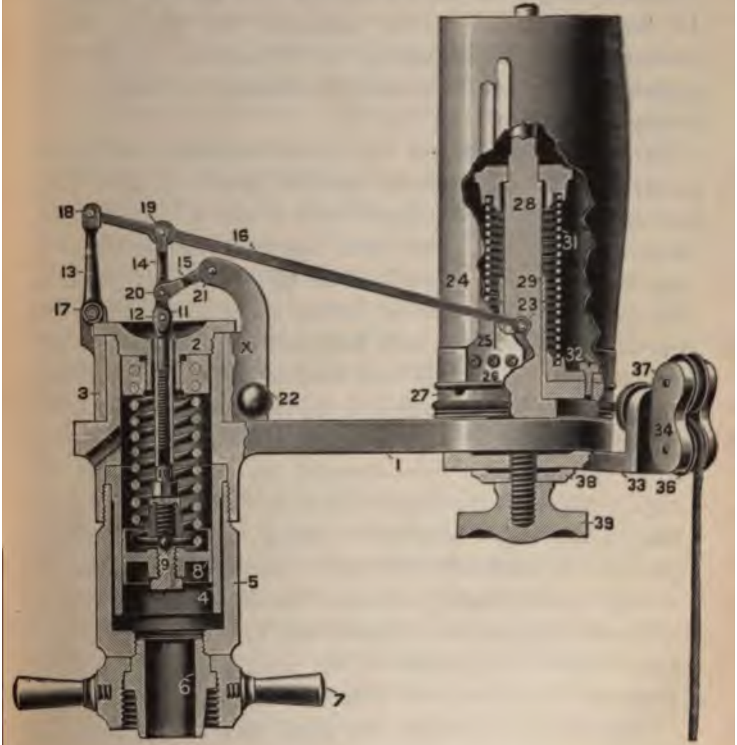
The illustration on page 19 shows the design and arrangement of its parts.

The cylinder, 4, in which the piston moves, is made of a special alloy, exactly suited to the varying temperatures to which it is subjected, and secures to the piston the same freedom of movement with high pressure steam as with low; and as its bottom end is free and out of contact with all other parts, its longitudinal expansion or contraction is unimpeded, and no distortion can possibly take place.

Between the cylinder, 4, and the casing, 5, is an annular chamber, which serves as a steam jacket; and being open at the bottom, can hold no water, but will always be filled with steam of nearly the same temperature as that in the cylinder.

The piston, 8, is formed from a solid piece of the finest tool steel. Its shell is made as thin as possible consistent with proper strength. It is hardened to prevent any reduction of its area by wearing, then ground and lapped to fit (to the twenty-thousandth part of an inch) a cylindrical gage of standard size. Shallow channels in its outer surface provide a steam packing, and the moisture and oil which they retain act as lubricants, and prevent undue leakage by the piston. The transverse web near its center supports a central socket, which projects both upward and downward; the upper part is threaded inside to receive the lower end of the piston-rod. The upper edge of this socket is formed to fit nicely into a circular channel in the under side of the shoulder of the piston-rod, when they are properly connected. It has a longitudinal slot, which permits the straight portion of wire at the bottom of the spring, with its bead, to drop to a concave bearing in the upper end of the piston-screw, 9, which is closely threaded into the lower part of the socket; the head of this screw is hexagonal, and may be turned with the hollow wrench which accompanies the indicator.

The piston-rod, 10, is of steel, and is made hollow for lightness. Its lower end is threaded to screw into the upper socket of the piston. Above the threaded portion is a shoulder having in its under side a circular channel formed to receive the upper edge of the socket, when these parts are connected together. When making this connection *be sure* that the piston-rod is screwed into the socket as far as it will go, that is, until the upper edge of the socket is brought firmly against the bottom of the channel in the piston-rod, before the piston-screw, 9, is tightened against the



bead at the foot of the spring. This is very important, as it insures a correct alignment of the parts and free movement of the piston within the cylinder.

The swivel head, 11, is threaded on its lower half to screw into the piston-rod more or less, according to the required height of the atmospheric line on the diagram. Its head is pivoted to the piston-rod link of the pencil mechanism. This adjustment of the position of the diagram upon the card is a valuable advantage peculiar to the Crosby indicator.

The cap, 2, rests on top of the cylinder, and holds the sleeve and all connected parts in place. It has a central depression in its upper surface, also a central hole, furnished with a hardened steel bushing, which serves as a very durable and sure guide to the piston-rod. It projects downward into the cylinder in two steps, having different lengths and diameters; both these and the hole have a common center. The lower and smaller projection is screw-threaded outside to engage with the like threads in the head of the spring, and hold it firmly in place. The upper and larger projection is screw-threaded on its lower half to engage with the light threads inside the cylinder; the upper half of this larger projection — being the smooth, vertical portion — is accurately fitted into a corresponding recess in the top of the cylinder, and forms thereby a guide by which all the moving parts are adjusted and kept in correct alignment, which is very important but practically impossible to secure by the use of screw threads alone.

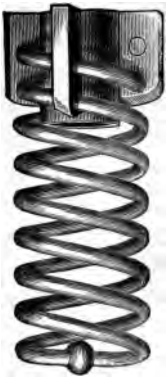
The sleeve, 3, surrounds the upper part of the cylinder in a recess formed for that purpose, and supports the pencil mechanism; the arm, X, is an integral part of it. It turns around freely, and is held in place by the cap. The handl for adjusting the pencil point is threaded through the a and being in contact with a stop-screw in the plat *be delicately adjusted* to the surface of th drum. It is made of hard wood with a lock-nut to maintain the adjustment.

The pencil mechanism is designed to afford sufficient strength and steadiness of movement, with the utmost lightness; thereby eliminating as far as possible the effect of momentum, which is especially troublesome in high speed work. Its fundamental kinematic principle is that of the pantograph. The fulcrum of the mechanism as a whole, the point attached to the piston-rod, and the pencil point are always in a straight line. This gives to the pencil point a movement exactly parallel with that of the piston. The mechanism is theoretically correct as well as mechanically accurate; the result is, therefore, mathematical precision in the pencil movement, not merely an approximation. The movement of the spring throughout its range bears a constant ratio to the force applied; and the amount of the movement of the piston is multiplied six times at the pencil point. The pencil lever, links, and pins are all made of hardened steel; the latter — slightly tapering — are ground and lapped to fit accurately, without perceptible friction or lost motion.

Springs. In order to obtain a correct diagram, the height of the pencil of the indicator must exactly represent in pounds per square inch the pressure on the piston of the steam engine at every point of the stroke; and the velocity of the surface of the drum must bear at every instant a constant ratio to the velocity of the piston. These two essential conditions have been attained to a greater degree of exactness in the Crosby indicator than in any other make, by a very ingenious construction and nice adaptation of both its piston and drum springs.

The piston spring is of unique and ingenious design, being made of a s[illegible] finest spring steel wire, [illegible] nd from the [illegible] coil, the ends of which [illegible] ed i[illegible] ng four radial wings drilled helically to receive and hold the spring securely in place.

Adjustment is made by screwing the ends into the head more or less, until exactly the right strength of spring is obtained, when they are there firmly fixed. This method of adjusting and fastening removes all danger of loosening coils, and obviates all necessity for grinding the wires — a practice fatal to accuracy in indicator springs.



The foot of the spring — in which freedom and lightness are of great importance, it being the part subject to the greatest movement — is a small steel bead, firmly "staked" on to the wire. This takes the place of the heavy brass foot used in other indicators, and reduces the inertia and momentum at this point to a minimum, whereby a great improvement is effected. This bead has its bearing in the center of the piston, and in connection with the lower end of the piston-rod and the upper end of the piston-screw, 9 (both of which are concaved to fit), it forms a ball and socket joint which allows the spring to yield to pressure from any direction without causing the piston to bind in the cylinder, which occurs when the spring and piston are rigidly united. Designing the spring so that any lateral movement it may receive when being compressed shall not be communicated to the piston and cause errors in the diagram, is of extreme importance. See also page 59.

The drum spring, 31, in the Crosby indicator is in form a helix, while in other indicators it is a long volute. It is obvious from the large contact surfaces of a long volute spring that its friction would be greater than that of a short, *open helical form* of like power; and that in a spring of this kind, for a given amount of compression—as in the movement of an indicator drum—the recoil will be greater and exerted more quickly in the helical than in the volute form.

If the conditions under which the drum spring operates be considered, it will readily be seen that at the beginning of the stroke, when the cord has all the resistance of the drum and spring to overcome, the spring should offer less resistance than at any other time; and at the beginning of the stroke in the opposite direction, when the spring has to overcome the inertia and friction of the drum, its energy or recoil should be greatest.

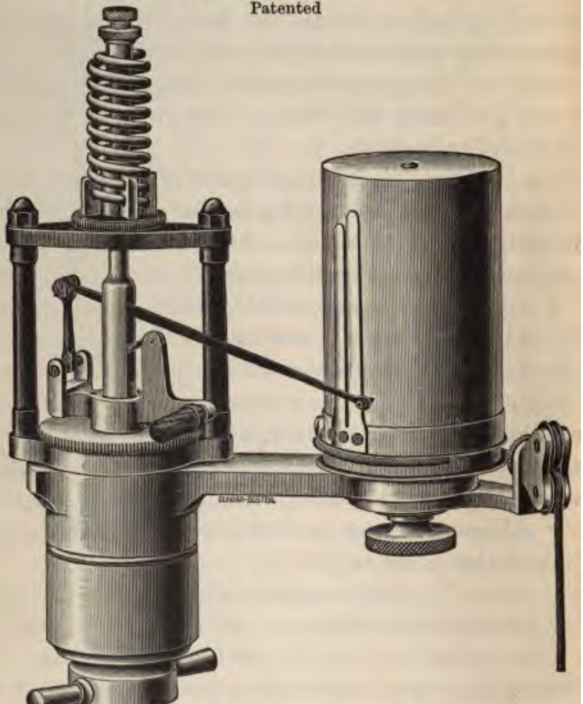
These conditions are fully met in all Crosby indicators, the drum spring being a helix having no friction, a quick recoil, and scientifically proportioned to the work it has to do. At the beginning of the forward stroke it offers to the cord only a very slight resistance, which gradually increases until at the end its maximum is reached. At the beginning of the stroke in the other direction, its recoil is greatest at the moment when it is most needed, and gradually decreases as the work it has to do decreases, until at the end of the stroke it is reduced to its minimum again. Thus, by a most ingenious balancing of opposing forces, the most nearly uniform stress on the cord is maintained throughout each revolution of the engine.

The drum, 24, and its appurtenances, except the drum spring, are similar in design and function to like parts of any indicator, and need not be particularly described. All the moving parts are designed to secure sufficient strength with the utmost lightness, by which the effect of inertia and momentum is reduced to the least possible amount. It is ordinarily 1½ inches in diameter, this being the correct size for high speed work, and answering equally well for low speeds. If, however, the indicator is to be used only for low speeds, and a longer diagram is preferred, it can be furnished with a 2 inch drum.

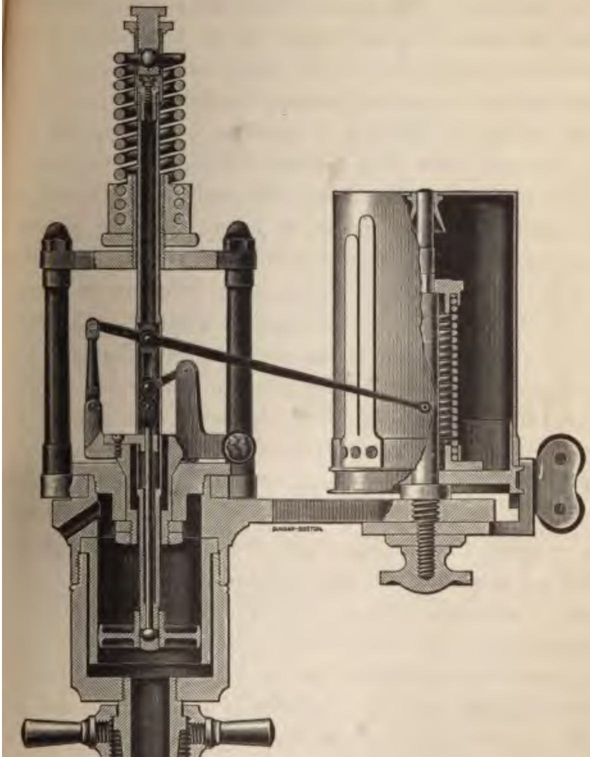
All Crosby indicators (except some of the Standard Ste Engine Indicators numbered below 3737) can be rea changed from right-hand to left-hand instruments as oc sion may require.

CROSBY NEW STEAM ENGINE INDICATOR

Patented



This instrument is a departure from the ordinary ste engine indicator. One difference is in the location of t spring, which is of the same form and construction as t one described and illustrated on page 22. This has be removed from the inside of the cylindrical case near t *piston to the* outside and affixed above the moving par where it will remain cool under all conditions of use. Whatever error arises from heat, therefore, as affecting the spring in the ordinary indicator, is not present in this instrument.



The other and more important difference lies in the size d shape of the piston. This piston is one square inch in ea, and is in form the central zone of a sphere. This creased area of the piston provides great active force with very light pencil mechanism. It is attached by a rod rectly *to the upper* part of the spring, and moves freely and without restraint notwithstanding there may be eccentricity in the action of the spring. In other words, this piston serves as a universal joint to take care of the torsional strains of the spring when it operates the pencil mechanism of the indicator. The pencil mechanism is connected to the piston by means of a rod having at its lower end a ball, which fits into a socket in the center of the piston. This socket is formed upon a headless slotted screw, adjustable in the piston, by which it is possible to take up all wear upon the ball joint that might develop after long service; but this adjustment should not ordinarily be disturbed, and care must always be taken to insure that the piston is firmly screwed to the piston-rod before the adjustment screw is tightened to just the amount sufficient to prevent any possibility of lost motion, without binding. The socket at the upper end of the piston-rod, to receive the ball bearing of the spring, is likewise upon an adjustable headless screw, which is independent of the screw beneath it that secures the swivel head of the piston-rod in its place.

The piston-rod moves freely within a sleeve attached to the base of the pencil mechanism, and, moving in a vertical line, compels the pencil to move also in a vertical line. Thus, any motion of the piston due to the movements of the spring, which causes the spring-rod to deviate, will not affect the pencil mechanism in its vertical course. The contact of the piston with the interior side of the cylinder is a line, and does not induce friction. Ordinarily, the piston of an indicator is a short cylinder fitted to slide easily within another cylinder. Such a piston is usually about one-half inch long, and in use will develop friction throughout its circumference. The piston so made must resist and overcome if possible the eccentricities of the spring in action. Yet, even then, there is always a want of freedom, notwithstanding there are devices to aid the piston in such case. This condition *tending to error is* recognized by engineers, and considered in the computations made of the diagram taken by the indicator. The freedom of the piston movement in this indicator dispenses with the necessity of this correction. When an indicator is to be used in highly superheated steam or in gases of high temperature this type, with outside spring, will give accurate results.

All parts of the indicator are constructed to give the greatest possible wear and durability with extreme lightness and freedom from all friction and the joints are upon hardened taper bearings. Means are thus provided to prevent all error or looseness, and no proper excuse exists for permitting any inaccuracy to develop. Although the adjustment is rarely needed and at every point is slight and simple, greater satisfaction will result if this work be done by skilled mechanics whose special experience enables them to work accurately and quickly, without the risk of damage involved in repairs undertaken by persons unfamiliar with such mechanism.

This indicator is made also for gas engine work with piston ½ square inch in area and special pencil mechanism; and may be made of steel when required, for ammonia.

The Crosby New Indicator appeals to the discriminating engineer because of its acknowledged superiority in design and workmanship, affording unapproachable accuracy in results. The linkage is a true parallel motion, and the relationship of the parts is not disturbed when changing the spring or cleaning the cylinder. The operation and adjustment of the indicator in use is simple and convenient.

On the following page examples are shown of its freedom of piston movement, being reproductions of the original test cards of a variety of springs.

CROSBY NEW INDICATOR NO. 2

Patented

This instrument has been designed to meet the demand *for an accurate and* trustworthy instrument of the outsideples of its action, continuously or intermittently at the will or convenience of the engineer. It furnishes recorded proof of such operation in a form that permits one diagram to be compared with another, and the variations during a cycle of operations to be intelligently observed in the sequence of their occurrence.

The examples on page 31 reproduced from actual diagrams taken on steam engines are given only to illustrate the foregoing statement. Single isolated diagrams can also be taken as with the ordinary drum.

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ASTOR, LENOX AND
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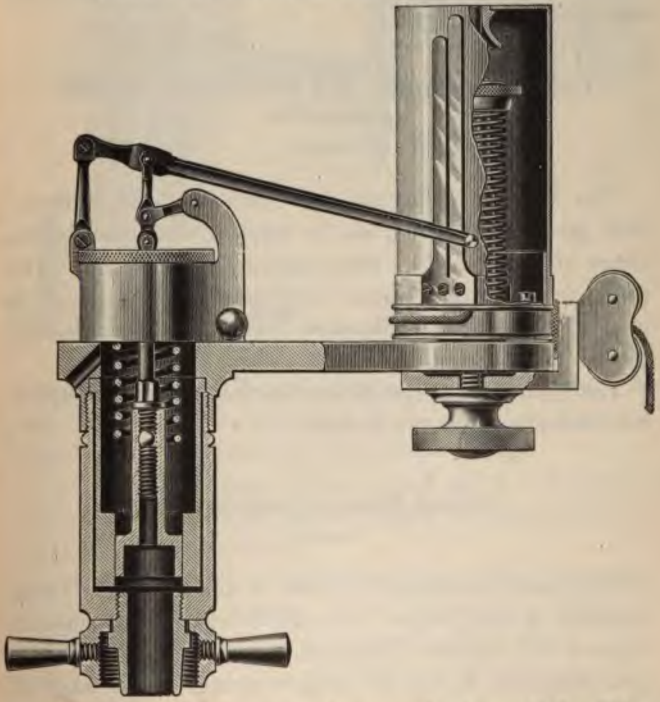
CHAPTER III

THE CROSBY GAS ENGINE INDICATOR. THE CROSBY COMBINED GAS AND STEAM ENGINE INDICATOR. THE CROSBY AMMONIA INDICATOR. THE CROSBY ORDNANCE INDICATOR. THE CROSBY HYDRAULIC INDICATOR

CROSBY STANDARD GAS ENGINE INDICATOR



The increase in the use of gas and of gasoline engines *has created a* demand for an accurate indicator capable of withstanding the heat, the high pressure, and the shock which result from the explosions in the cylinder.

The Crosby Standard Gas Engine Indicator designed to meet these requirements has given perfect satisfaction. The piston is ¼ of a square inch in area and springs made for ½ inch pistons have their rating doubled when used in this instrument. Its great accuracy and durability have been fully demonstrated, and its method of construction leads to the least error in the taking of such diagrams.

The Crosby New Indicator is made also for gas engine work, with piston ¼ square inch in area, of the design described on page 25.

CROSBY COMBINED GAS AND STEAM ENGINE INDICATOR

Patented

This indicator is for use either with the steam engine or with gas or oil engines, and is supplied with two pistons, either of which can be fitted separately as desired. The piston for steam is ½ square inch in area, the same as is used with the Crosby Standard Steam Engine Indicator; the one for gas or oil is ¼ square inch in area.

The cylinder is of special construction to suit the respective diameters of the two pistons.

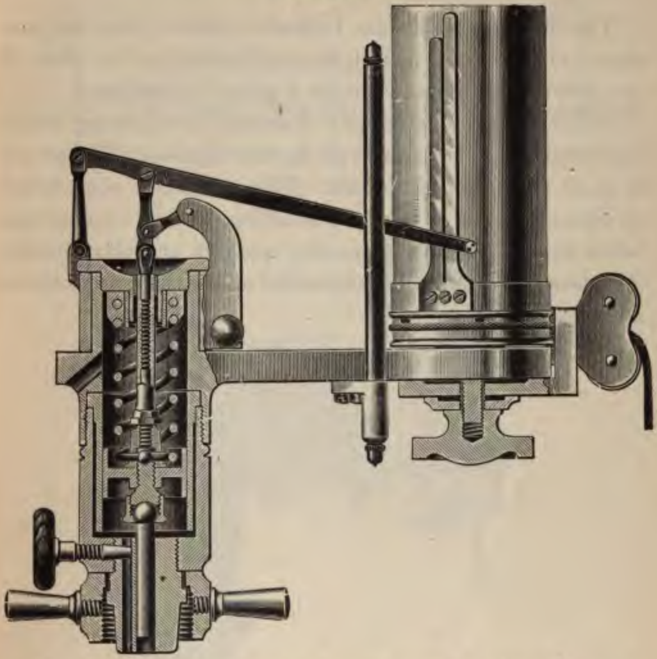
CROSBY AMMONIA INDICATOR

Patented

The Crosby Ammonia Indicator is made like the Crosby Standard Steam Engine Indicator, except that all surfaces exposed to the action of ammonia are of steel. The Crosby New Indicator, having the outside spring, is also made of steel when required for ammonia. For ammonia indicators, *both common and* three-way cocks are made of steel.

CROSBY ORDNANCE INDICATOR

Patented



This instrument will give a true record of high pressures, such as obtain in the operation of the pneumatic gun carriage for heavy ordnance, or in hydraulic pumps.

The pencil mechanism is strong, and has a post bearing lightly against the pencil arm to keep it in contact with the drum during sudden shocks.

The piston is $\frac{1}{40}$ of a square inch in area, and is fitted into a cylinder at the bottom of the instrument. There is a by-pass by which the pressure may be transmitted to the larger piston above, when the pressures to be recorded are not too high for the capacity of the spring. This by-pass is closed by a cock when the small piston is to be used.

CROSBY HYDRAULIC INDICATOR

Patented

The Crosby Hydraulic Indicator differs from the one shown in the cut in that it has no by-pass and in place of the piston in the upper chamber a guide is substituted.

It is a strong and efficient instrument for indicating under high pressure conditions in all liquids or gases. The piston is $\frac{1}{40}$ of a square inch in area. The cylinder is constructed in such a manner as to afford a uniform area of cross-section below the piston, thus preventing pockets or enlargements. The pencil mechanism is substantial and without appreciable inertia effect.

CHAPTER IV

INDICATOR ATTACHMENTS AND ACCESSORY APPARATUS

SARGENT IMPROVED ELECTRICAL ATTACHMENT

For Steam or Gas Engine Indicators

Patented

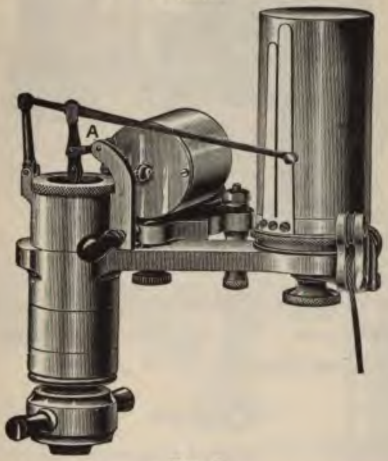


FIG. 3

In making elaborate tests of power plants, it has heretofore been necessary to employ as many assistants as there were indicators used, but the difficulty of securing simultaneous action on their part is so great that satisfactory work is rarely to be obtained, and more certain means to that end are now considered necessary.

Mr. Frederick Sargent, M.E., invented and patented an electrical device applicable to an indicator, by means of which any number of instruments can be operated and di grams taken at the same instant of time, simply by closi an *electric circuit.* We are the sole owners of this pa and of the rights under it; and are the sole makers of this apparatus, which has been modified and improved.

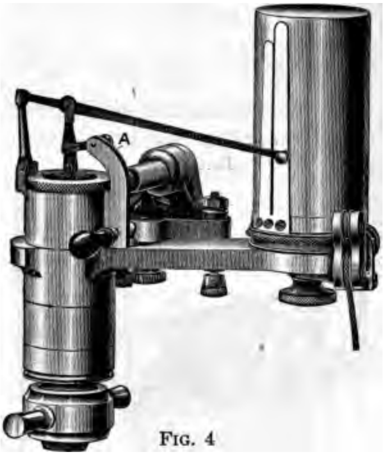


FIG. 4

Fig. 3 shows a Crosby Standard Indicator fitted with this electrical attachment. Fig. 4 shows the same indicator fitted with a Circuit Closer.

Description

Fig. 5 represents the Sargent Improved Electrical Attachment, consisting of an electromagnet, A, which is supported by a bracket, B, which also secures it to the indicator plate. Binding posts, C, C, are attached to the same bracket. The armature D is opposed to the magnet by a spiral spring in the center of the coil, the tension of which is adjustable by means of the screw E, at the back of the magnet. The movement of the armature outwardly is limited by two screws, 1 and 2. To the armature is secured a small latch or hook, F, which is free to work vertically, and engage with the arm A, Fig. 3. The thumb-screw G is for fastening the attachment to the plate of an indicator through a *hole therein.*

Fig. 6 represents the Circuit Closer, and is designed to operate the electrically connected indicators, by closing the circuit through them whenever the stylus or marking point is put against the paper on the drum of the indicator to which it is attached. This enables the engineer making the test to control this indicator directly by hand — a feature often desirable — and by its use one Sargent attachment is dispensed with.

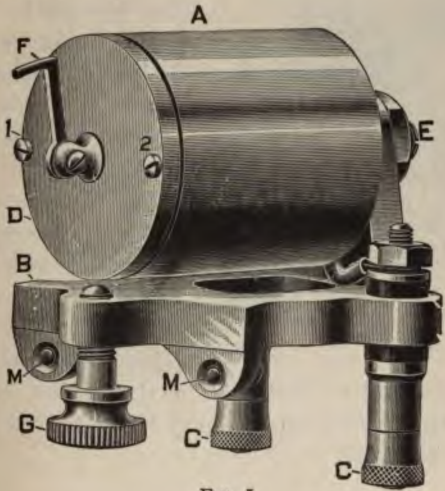


FIG. 5

It consists of a bracket, H, with a tubular projection, I, fastened to it which contains the circuit closing mechanism. It is attached to the indicator plate by the thumb-screw, J, in precisely the same way that the magnets are to the other indicators, and is electrically connected in the same manner through the binding posts, K, K.

To Attach the Sargent Improved Electrical Attachment

To get the position of the hole in the frame of the indicator, take out the screw G (Fig. 5), and place the bracket holding the magnet against the plate of the indicator, so that the hook F (Fig. 5), when placed horizontally, will point to the middle of the arm A (Fig. 3); then scribe through the screw-hole its location upon the plate of the indicator; remove the attachment, and drill a hole where marked that will allow the screw G (Fig. 5) to pass through it. Screw the attachment to the plate and adjust it so that there will be no looseness, turning the screws M M (Figs. 5 and 6), setting them up gently.

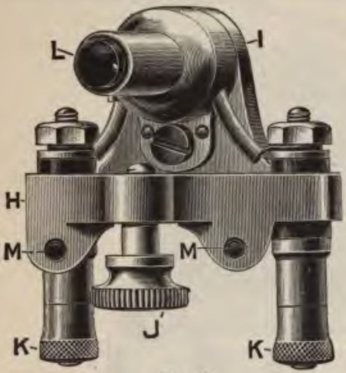


FIG. 6

Drop the hook F (Fig. 5) to a horizontal position, and bring the arm A (Fig. 3) up to its working position, and mark on it the center of the hole to be drilled for a screw-eye. This hole should be so drilled that the latch will stand level with the plate when in use. The size of the hole may be determined from the screw-eye furnished.

To Attach the Circuit Closer

The position of the hole in the indicator plate for attaching the Circuit Closer, Fig. 6, is determined in the same manner as for the electromagnet, taking care that the button L (Fig. 6) impinges the center of the arm A (Fig. 4) when the sleeve is turned into the correct position for use.

The sleeve handle of the indicator is unscrewed far enough to allow the button L (Fig. 6) in the end of the projection I to go in as far as it will, then the marking point must be adjusted until it makes the desired tracing on *the paper.*

To Operate the Sargent Improved Electrical Attachment and Circuit Closer

For the purpose of illustrating the manner of operating the attachment, assume that it is desirable to procure simultaneous diagrams from a compound engine, taking cards from the ends of each cylinder. Attach the indicators to the engine and arrange the drum motion in the usual manner. On each indicator secure the electrical attachment to its plate by means of screw G, as above described. Make the connections with the battery, having all of the several magnets and the circuit closer in series. Place the paper upon the drum, and bring the pencil arm into such a position as will allow the latch F to drop into the screw-eye before mentioned.

Press the armature firmly against the magnet, and adjust the marking point to the paper in the usual manner. The sleeve handle must be unscrewed enough to allow the full operation of the armature. The circuit should be closed and the armature tension springs adjusted, so that the connected attachments will work simultaneously. Everything should now be in readiness to take diagrams. Connect the drum motions, open the indicator cocks, and as soon as desirable close the circuit, and instantly all of the pencils will be brought against the papers and will remain there as long as the circuit is kept closed.

In order to put on new papers, disengage the drum motions, lift the latch, and swing the pencil arm out of the way.

The Electric Battery

The amount of battery power required will vary with circumstances, and will range from one to two or more cells of a No. 2 Samson battery, or its equivalent.

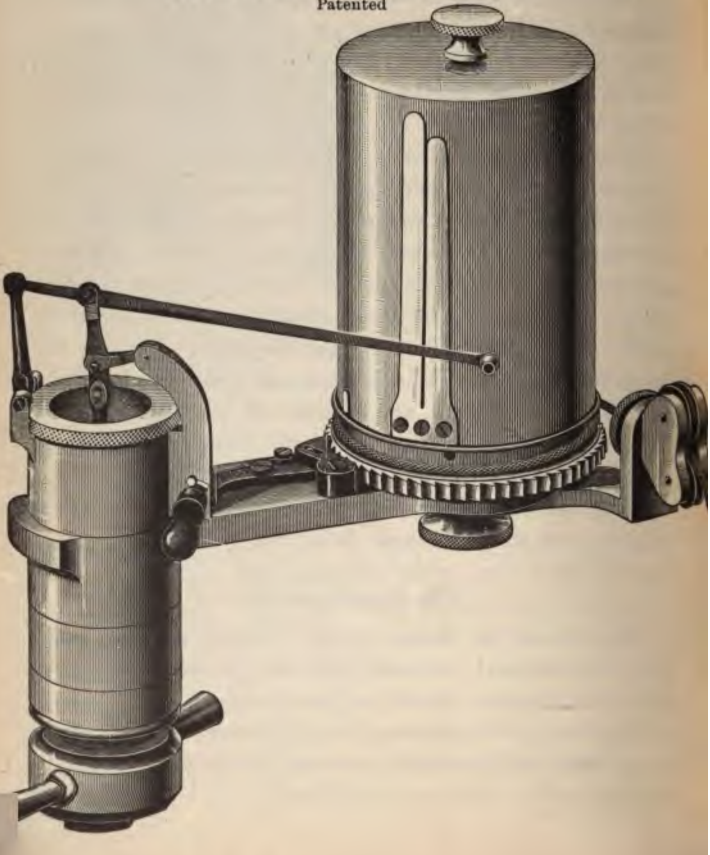
The battery for operating the attachments is enclosed in a neat hardwood box with a suitable handle for carrying it, and is *sealed so as to* prevent slopping. It is very compac and portable, being at the same time extremely active, long lived, and especially adapted to open circuit work.

The connections to the indicator attachments can be made with the battery without opening the box, the binding posts being on the outside.

This battery, with a quantity of suitable wire for making connections, is furnished with the attachment.

CROSBY STANDARD STEAM ENGINE INDICATOR WITH 2 INCH DRUM AND DETENT

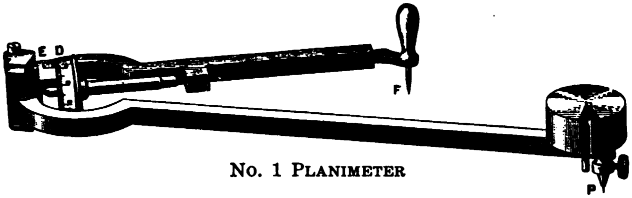
Patented



The detent device has its pawl attached to the base plate of the indicator and is provided with a suitable handle for operating it. The pawl engages the ratchet located at the base of the drum. See also page 73.

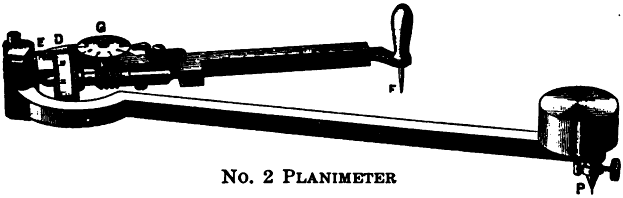
PLANIMETERS

With Directions for Using the Planimeter on Indicator Diagrams



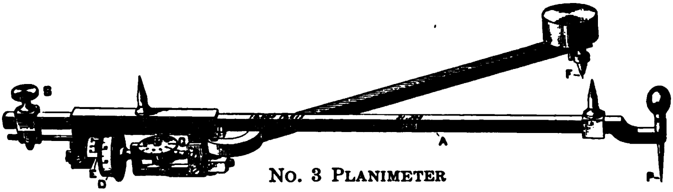
No. 1 Planimeter

This cut represents the No. 1 planimeter. It is the simplest form of the instrument, having but one wheel, and is designed to measure areas in square inches and decimals of a square inch. The figures on the roller wheel D represent *units*, the graduations on the wheel represent *tenths*, and the vernier gives the *hundredths*.



No. 2 Planimeter

This cut represents the No. 2 planimeter, which is the same as the No. 1, with the addition of a counting disc G, the figures on which represent *tens* and mark complete revolutions of the roller wheel. By this means areas greater than ten square inches can be measured with facility. The result is given in square inches and decimals, and the reading from the roller wheel and vernier is the same as with No. 1.



No. 3 Planimeter

The No. 3 planimeter differs somewhat in design from the two previously described. It is capable of measuring larger areas, and by means of the adjustable arm A, giving the results in various denominations of value, such as square decimeters, square feet, and square inches; also of giving the average height of an indicator diagram in fortieths of an inch, which makes it a very useful instrument in connection with indicator work.

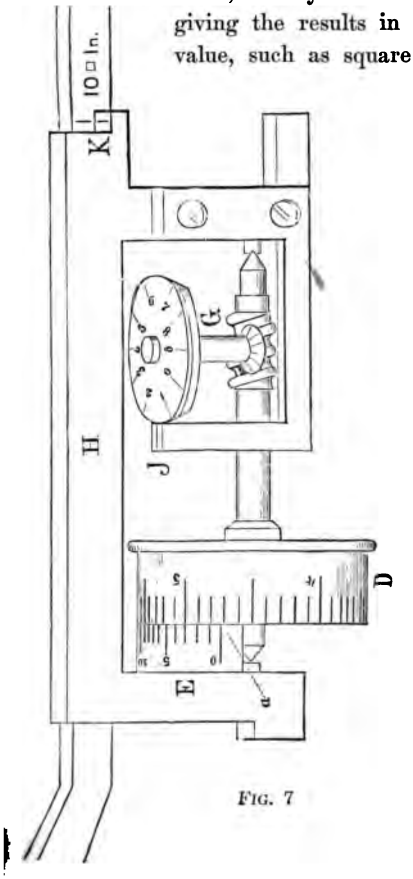


Fig. 7

Recording Mechanism

Fig. 7 shows in detail the recording mechanism of a No. 3 planimeter, from which the method of reading from either instrument may be easily understood. G is the counting disc, D the roller wheel, and E the vernier. From the counting disc we read 1 (ten), for the last figure that has passed the index line on the post J; from the roller wheel we read 4 (units), for the last figure that has passed zero on the vernier; we also read 7 (tenths), for that number of graduations beyond 4 that have also passed zero on the vernier (shown by the dotted line *a*), then from the vernier we read 3 (hundredths), because the *third* graduation on the vernier *coincides* with a graduation on the roller wheel.

The complete reading will then be 14.73 square inches.

When starting from zero the movement of the counting disc need not be noted when measuring *single* indicator diagrams, as they are of less than ten square inches area.

Directions for Measuring an Indicator Diagram with a No. 1 or No. 2 Planimeter

Care should be taken to have a flat, even, unglazed surface for the roller wheel to travel upon. A sheet of dull finished cardboard serves the purpose very well.

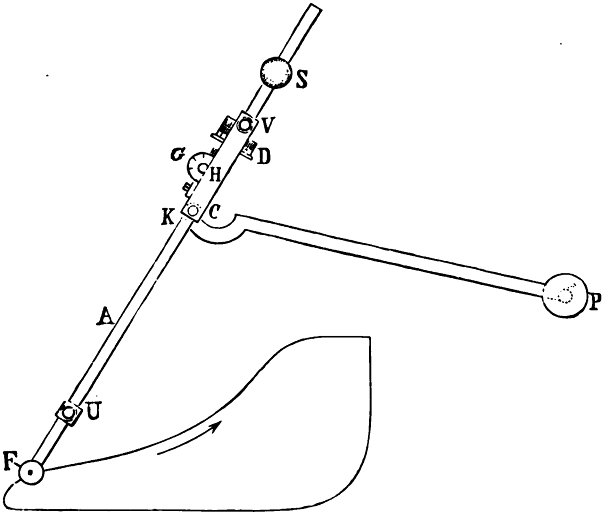


FIG. 8

Set the weight in position on the pivot end of the bar P, and after placing the instrument and the diagram in about the position shown in the cut (Fig. 8), press down the needle point so that it will hold its place; set the tracer point at any given point in the outline of the diagram, as at F, and adjust the roller wheel to zero. Now follow the outline of the diagram carefully with the tracer point, moving it in the direction indicated by the arrow, or that of the hands of a watch, until it returns to the point of beginning. The result may then be read as follows: Suppose we find that the largest figure on the roller wheel D, that has passed by zero on the vernier E, to be 2 (units), and the number of graduations that have also passed zero on the vernier to be 4 (tenths), and the number of the graduation on the vernier which exactly coincides with the graduation on the wheel to be 8 (hundredths), then we have 2.48 square inches as the area of the diagram. Divide this by the length of the diagram, which we will call 3 inches, and we have .8266 inches as the *average height* of the diagram. Multiply this by the scale of the spring used in taking the diagram, which in this case is 40, and we have 33.06 pounds as the *mean effective pressure* per square inch on the piston of the engine.

Directions for Using the No. 3 Planimeter

No. 3 planimeter is somewhat differently manipulated, although the same general principle pertains. The figures on the wheels may represent different quantities and values according to the particular adjustment of the sliding arm A. If it is desired merely to find the area in square inches of an indicator diagram, set the sliding arm so that the 10□ inch mark will exactly coincide with the vertical mark on the inner end of the sleeve H at K, Fig. 7. The sliding arm is released or made fast by means of the set-screw S.

With the wheels at zero and the planimeter and diagram in the proper position (shown in Fig. 8) trace the outline carefully, and read the result from the roller wheel and vernier, the same as directed for the No. 1 and No. 2 instruments.

Example: Suppose, in a diagram so measured, we read from the figures on the roller wheel 3 (units), from the graduations on the roller wheel 0 (tenths), and from the vernier 8 (hundredths), then we have an area of 3.08 square inches; to find the average height we divide this by its length, which is 3½ inches. Thus, 3.08 ÷ 3.5 = 0.88 of an inch, the average height. This multiplied by the scale of the spring used, which in this case is 60 pounds, gives 52.8 pounds as the M. E. P.

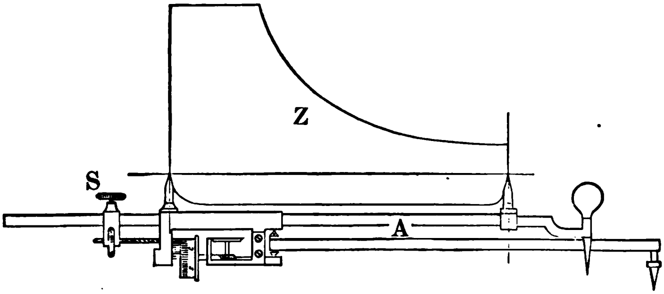


FIG. 9

To find the average height of an indicator diagram at one measurement, set the sliding arm so that the steel points on its upper side shall be just the length of the diagram — measured on a line parallel with the atmospheric line — apart, as shown in Fig. 9. With this adjustment the figures on the counting disc represent hundreds, those on the roller wheel tens, the intermediate graduations units, and the vernier gives the decimal. Place the instrument in position with the wheels at zero, and trace the outline of the diagram. The result of the reading will be its *average height in fortieths of an inch.*

are filled with cold water and that they do not leak; open the valve E wide, and wait ten or fifteen minutes till the whole apparatus is heated. Read the gage B and add the pressure of the atmosphere* to get the absolute pressure in the calorimeter; find the corresponding temperature from a table of the properties of saturated steam and compare with the temperature in the calorimeter given by the thermometer; the excess of the latter over the former is the superheating of the steam in the calorimeter. The flow of steam through the calorimeter will be sufficient to make the loss by radiation of no consequence and no correction need be applied.

When all is ready, read the pressure of the steam p in the main steam pipe, the temperature t_s in the calorimeter, the pressure $p_{/}$ in the calorimeter, and take the pressure p_a of the atmosphere. From a table of the properties of saturated steam, find the temperature $t_{/}$ corresponding to the absolute pressure $P_{/} = p_{/} + p_a$

From the same tables find the total heat $\lambda_{/}$ corresponding to the pressure $P_{/}$; also the heat of vaporization r and the heat of the liquid q corresponding to the absolute pressure in the steam pipe $P = p + p_a$

The weight or moisture in 1 pound of moist steam drawn from the steam pipe is to be calculated by the equation

$$\text{Priming} = 1 - \frac{\lambda_{/} + 0.48\,(t_s - t_{/}) - q}{r}$$

in which the factor 0.48 is the specific heat of superheated steam at constant pressure.

The calculation will be readily understood from the following example:

Pressure in steam pipe, $p = 69.8$ pounds.

Pressure in the calorimeter, $p_{/} = 12$ pounds.

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* NOTE. · The pressure of the atmosphere is commonly assumed to be 14.7 pounds per square inch; it may be taken by aid of a barometer or obtained from published records of the Weather Bureau for the day. Inches of mercury can be reduced to pounds per square inch by multiplying by 0.49.

Pressure of the atmosphere, $p_a = 14.8$ pounds.

Temperature in the calorimeter, $t_s = 268.2°$ F.

Absolute pressure in steam pipe, $P = p + p_a = 69.8 + 14.8 = 84.6$ pounds.

Absolute pressure in calorimeter, $P_{,} = p_{,} + p_a = 12 + 14.8 = 26.8$ pounds.

Temperature of saturated steam at 26.8 pounds, 243.9° F.

Total heat at 26.8 pounds, $\lambda_{,} = 1161.4$ thermal units.

Heat of vaporization at 84.6 pounds, $r = 896.8$ thermal units.

Heat of the liquid at 84.6 pounds, $q = 286.2$ thermal units.

$$\text{Priming} = 1 - \frac{1161.4 + 0.48\ (268.2 - 243.9) - 286.2}{896.8} = 0.002$$

a result that is commonly stated as $\frac{2}{10}$ per cent priming.

Steam delivered by a boiler or supplied to an engine commonly contains a small amount of moisture, but if the steam is very wet from any cause it may fail to superheat in the calorimeter and in such case the calorimeter cannot be used for determining its quality. Should this occur in a boiler test, it indicates either that the design of the boiler is defective or that it is in bad condition and needs cleaning. The steam supplied to an engine may be deprived of the greater part of its moisture, if it be very wet, by passing it through a separator. It has been found that steam used in good ordinary practice will always superheat in the throttling calorimeter.

While it is advisable that the gages and thermometer used with the throttling calorimeter should be of first quality and entirely reliable, the errors that such instruments are liable to have do not have a serious effect on the result. Thus, at 100 pounds absolute and with atmospheric pressure in the calorimeter, 10° F. superheating indicates 0.035 priming; should the thermometer be wrong 5° F. and indicate 15° *F., the priming* will appear to be 0.032. In a insert the combined wrench and piston-rod until the bead of the spring rests in the concaved end of the latter; then invert the piston and pass the transverse wire at the bottom of the spring through the slot until the threads at the bottom of the piston-rod engage those inside the socket of the piston, and with the wrench screw it in as far as it will go; that is, until the upper edge of the socket is in contact with the bottom of the channel in the shoulder of the piston-rod. The piston-screw should be loosened slightly before this last operation and afterwards set up against the bead lightly, to provide against any lost motion, yet not so as to make it rigid. Next, hold the sleeve and cap in an upright position — so that the pencil lever will drop to its lowest point — and engage the threads of the swivel head with those inside the piston-rod and screw it up until the threads on the lower projection of the cap engage those in the spring head, and continue the process until the latter is screwed firmly up against the cap. Then, letting the cap go free and holding only by the sleeve, continue to turn the piston (together with its connections) until the top of the piston-rod is flush with the shoulder on the swivel head.

The piston and its connections may now be inserted in the cylinder and the cap screwed down, which will carry all parts into their proper places.

To detach a spring simply reverse this process.

To change the location of the atmospheric line of the diagram. First, unscrew the cap and lift the sleeve, with its connections, from the cylinder; then — holding the sleeve with the left hand — with the right hand turn the piston and connected parts towards the left, and the pencil point will be raised, or to the right and it will be lowered. One complete revolution of the piston will raise or lower the pencil point $\frac{1}{8}$ inch and this should be the guide for whatever amount of elevation or depression of the atmospheric line is needed.

To change to a left-hand instrument. If it is desired to make this change: First, remove the drum shell from its base by a straight upward pull; then, with the hollow wrench, remove the hexagonal stop-screw in the drum base, and screw it into the vacant hole marked L; next, *reverse* the position of the adjusting handle in the arm; also, the position of the metallic point in the pencil lever; then replace the drum and the change from right to left will be completed.

This applies to all indicators except Standard Indicators numbered below 3737.

The tension on the drum spring may be increased or diminished according to the speed of the engine on which the instrument is to be used, as follows: Remove the drum shell by a straight upward pull; then raise the *head* of the spring *above* the *square* part of the spindle and turn it to the right for more or to the left for less tension, as required; then replace the head on the spindle.

Before attaching the indicator to an engine, be sure to *blow steam freely through the pipes and cock* to remove any particles of dust or grit that may have lodged in them.

After using the indicator it should be carefully wiped and oiled. For this purpose it is not often necessary to disturb the paper drum, but the cylinder cap should be unscrewed and all the connected parts lifted out; then the piston, piston-rod, and spring should be detached and all carefully wiped with cloth or tissue paper until perfectly dry, then slightly oiled with a lubricant of good quality; the inside of the cylinder should also be oiled. After this is done the piston and piston-rod should be replaced in the cylinder, but the spring should be kept on its stud in the box when the instrument is not in use. The box should be kept in a dry place.

After the indicator has been unused for any length of time the oil used at its last cleaning may have become gritty or gummy; it should be wiped off with a soft cloth or tissue paper saturated with naphtha or benzine, and then freshly oiled before it is used. This keeps the instrument in prime condition and insures the best results. An occasional naphtha bath will cleanse every part; but oil should be thoroughly applied afterwards to prevent corrosion.

If any grit or other obstruction gets into the cylinder it may score the cylinder wall and also cause friction and sticking of the piston. This will seriously affect the diagram and lead to bad results. It is not difficult to detect such trouble and it should be remedied at once by taking out the piston, detaching the parts and cleaning them as above described, when the disturbing cause will generally be removed. The inner wall of the cylinder should be frequently lubricated.

It is essential to know whether or not the indicator is in good condition for use; especially, to know that the piston has perfect freedom of motion and is unobstructed by undue friction. To test this successfully, detach the spring and afterwards replace the piston and piston-rod in their usual position, then, holding the indicator in an upright position by the cylinder, in the left hand, raise the pencil arm to its highest point with the right hand and let it drop. It should freely descend to its lowest point. If the opening at the bottom of the indicator cylinder be closed, by placing the thumb over it, the piston should descend slowly from its highest point, if there is no excessive clearance or leakage past the piston. These tests should be made only when all the parts are warm from the steam, in the condition as actually used, and the piston and cylinder should be carefully wiped and lubricated beforehand, at the time when the spring is removed.

The pencil should always have a smooth, fine point; a *fine file is the best* instrument to use to sharpen it.

The Crosby New Steam Engine Indicator

To remove the piston from the cylinder, unscrew the cap and lift it with the plate supporting the posts and spring from the indicator.

The attachment of the spring is easily and quickly made. Remove the knurled nut above the bead and release the binding nut below the spring head and then unscrew the spring from the threaded bushing of the cross-bar to which it is attached.

When the spring is in place, the pencil may be adjusted to any desired position on the drum by loosening the binding nut below the spring head and screwing the spring upward or downward, carrying with it the pencil mechanism. So located, with the binding nut again screwed firmly into place to lock the spring, the pencil is held in position.

The suggestions and advice as to the care of the Standard Steam Engine Indicator apply equally well to the Crosby New Indicator.

INDICATOR SPRINGS

Indicator springs are made of different sizes of steel wire, to adapt them, in point of strength, to different pressures of steam.

To adapt the indicator to any steam pressure, springs are made to the following scales:

4 8 12 16 20 30 40 50 60 80 100 120 150 180

The number stamped on the spring represents the number of pounds pressure to the square inch which are required to compress it sufficiently to move the pencil vertically 1 inch on the diagram. The strength of the spring that should be used for indicating an engine depends upon the maximum steam pressure to which it will be subjected; it should be of such a strength or number that the diagram will not be *over 1¾ inches* high. The proper spring to be us in any given case may be found as follows: Divide the boiler pressure, expressed in pounds, by the desired height of the diagram, expressed in inches, and the result will be the number of the spring required. For instance, if the boiler pressure is 70 pounds and the desired height of the diagram is 1¾ inches, then 70 ÷ 1¾ = 40, the number of the spring required.

In practice, the best diagram for measuring and interpreting is one in which the length is not more than twice the height, for the reason that the points of cut-off, exhaust and compression are better defined than in a longer diagram.

The late John C. Hoadley, an eminent authority in the use of the steam engine indicator, said:

"There are good reasons for keeping the diagram of very moderate length. From 2½ to 3 inches will be found long enough to admit of all useful division, and the movement of the paper cylinder will be slower, and the tracing correspondingly more delicate than if a longer card is made. A similar remark applies equally well to the vertical motion, which can be reduced to any amount desired by using springs of suitable stiffness."

Sir Frederick Bramwell succeeded in obtaining very satisfactory diagrams at extraordinary speeds and high pressures, by limiting the dimensions to one inch in length and width, but this was an extreme case. It is generally more convenient and satisfactory to make the length about 2½ to 3½ inches, and the height from the atmospheric line about 1½ to 1¾ inches, by selecting a spring adapted to the pressure.

INDICATOR SCALES

The scales for measuring diagrams are sometimes made of steel, with several different graduations marked on each; *they are more often* made of boxwood, and these we recommend as being more easily read and less liable to be misapplied, as there is only one graduation marked on each scale of this kind. The scale with which to measure the height of a diagram must correspond with the number of the spring used in the indicator when the diagram is taken. For a diagram taken with a number 40 spring, use a scale graduated in 40 divisions to the inch, or for a diagram taken with a number 60 spring, use a scale graduated in 60 divisions to the inch, and so of all other springs.

Place the scale on the diagram at right angles to the atmospheric line with the zero mark of the scale exactly on that line; the figures set against these divisions show — at whatever point the line of the diagram crosses the edge of the scale — the pressure per square inch exerted by the steam on the piston of the indicator in tracing it. The divisions below the zero point show vacuum.

The most common scales are those numbered 40, 50, and 60; that is, an inch of vertical height on the scale represents, according to the number of the corresponding spring used, 40, 50, or 60 pounds of steam pressure per square inch in the cylinder.

CHAPTER VI

HOW AND WHERE TO ATTACH THE INDICATOR

The indicator should be attached close to the cylinder whenever practicable, especially on high speed engines. If pipes must be used they should not be smaller than half an inch in diameter and as short and direct as possible; if long pipes are needed they should be slightly larger than half an inch and covered with a non-conducting material.

Diagrams should be taken from both ends of the cylinder of an engine. If the diagram from one end is satisfactory it is not safe to assume that one taken at the other end will be equally so; it is often otherwise, owing to the varying conditions usually found. The lengths of thoroughfares, the points of valve opening and closing, and the lead, are variable and should be carefully adjusted to secure the best results, and this can only be done through the instrumentality of an indicator.

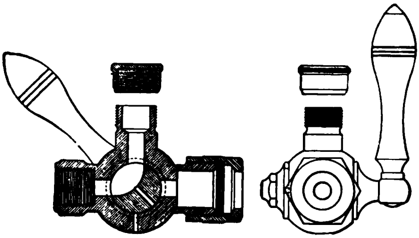


FIG. 12

When only one indicator is employed, it is generally attached to a three-way cock (Fig. 12), which is located midway in the line of pipe connecting the holes at either end of the cylinder; by this arrangement diagrams can be *taken from either* end simply by turning the handle of the three-way cock. In such a case, the second diagram should be taken as quickly as possible after the first, so as to be under like conditions of speed, pressure, and load.

The indicator *can* be used in a horizontal position, but it is more convenient to take diagrams when it is in a vertical position, and this can generally be obtained, when attaching to a vertical engine, by using a short pipe with a quarter upward bend. No putty or red lead should be used in making any joints, as particles of it may be carried by the steam into the indicator, and great harm result therefrom; if a screw fits loosely, wind into the threads a little cotton waste, which will make a steam-tight joint. The indicator should never be set so as to communicate with thoroughfares where a current of steam will flow past the orifice leading to the indicator, as the diagrams taken under such conditions would be of no practical value.

The cylinders of most modern steam engines are drilled and tapped for the indicator and have plugs screwed into the holes, which can readily be removed and the proper indicator connections inserted. But when this is not the case, the engineer should be competent to do it under the directions here given.

When drilling holes in the cylinder the heads should be removed if convenient, so that one may know the exact position of the piston, the size of ports and passages, and be able to remove every chip or particle of grit which might otherwise do harm in the cylinder or be carried into the indicator and injure it. When the heads cannot be taken off, it can be arranged so that a little steam may be let into the cylinder, when the drill has nearly penetrated its shell, so that the chips may be blown outward — care being taken not to scald the operator.

Each end of the cylinder should be drilled and tapped for ½-inch pipe thread. The holes must always be drilled into the *clearance space*, at points beyond the range o piston when at the end of the stroke, so as not to be obstructed by it, and away from steam passages, to avoid strong currents of steam. By placing the engine on a dead center, it is easy to tell how much clearance there is, and the hole should be drilled into the middle of this space; the same process should be repeated at the other end of the cylinder.

On horizontal engines the most common practice is to drill and tap holes in the *side* of the cylinder at each end and insert short ½-inch pipes with quarter upward bends, into which the indicator cocks may be screwed; on some horizontal engines it may be more convenient to drill and tap into the top of the cylinder at each end and screw the cocks directly into the holes. On vertical engines, for the upper end of the cylinder the cock may be screwed into the upper *head* or *cover*, and for the lower end, into the *side* of the cylinder, after drilling and tapping the necessary hole. It is preferable to drill the holes in the *sides* of a cylinder rather than the heads, because the former gives better results and requires less pipe and fittings.

Before deciding just where to drill the holes it is wise to consider *all* the conditions of the case and devise the *whole* plan for indicating the engine.

Sometimes a drum motion can be erected more advantageously in one place or position in the engine room than another, or one kind may be better adapted for a given place than another. Again, the type of engine and position of the steam chest, the kind of cross-head and the best means for attaching to it, the position of the eccentric, its rods and connections, all should be taken into account when determining the best places to drill the cylinder and locate the indicator, in order to secure a proper connection with the reducing motion, a perfectly free passage for steam to the indicator and the most convenient access to the instrument for taking diagrams.

CHAPTER VII

DRUM MOTION

The motion of the paper drum may be derived from any part of the engine which has a movement coincident with that of the piston. In general practice and in a large majority of cases, the cross-head is chosen as being the most reliable and convenient.

·The movement of the cross-head, whatever it actually is, must, by appropriate mechanism, be reduced to the length of the diagram to be taken, and this reduced motion must be, in point of rapidity, in exact ratio to the motion of the piston.

To obtain this reduced motion, various devices may be employed.

The reducing lever in some one of its various forms can be easily made and adapted to suit almost any conditions.

A common form of this device is shown in Fig. 13, and answers fairly well for large and quick running engines. It should be made of straight grained pine, one inch or more in thickness, about three inches wide at the top, and tapering to a width of about two inches at the bottom; its length should be at least one and a half times the length of the stroke of the piston.

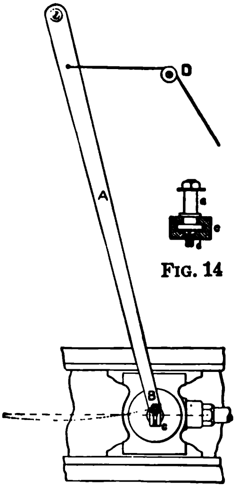


FIG. 14

FIG. 13

The lever, A, is suspended by a bolt from the ceiling or from a truss *or frame* overhead, prepared for that purpose, in such a manner as to permit it to swing edgewise and parallel with the guides of the engine. Near the bottom of the lever is a steel stud, secured by a nut on the outside shown at B. This stud has a T-head projecting inwardly from the lever, and is formed to run freely, but without looseness, in a T-slot, cut in an iron plate, and firmly attached to the center of the cross-head, which, as it moves to and fro, gives to the lever the necessary swinging motion.

Fig. 14 shows the arrangement of the T-headed stud, *a*, in connection with the slotted iron plate, *c;* one of the screws by which it is attached to the cross-head is shown at *d*. The head of the stud should be about one inch in diameter and the shank about one-half inch. The T-slot is milled out of a cast-iron plate of suitable size and shape to give the proper run for the stud.

When the lever is perpendicular, or in the middle of its path, the stud should be near the bottom of the slot, which should be long enough to retain the stud when the cross-head is at the extreme end of the stroke.

By this device the bottom end of the lever is moved the full distance that the cross-head travels in either direction, and for this reason it is more accurate than a lever of the same kind having its lower end slotted to work on a stud inserted in the cross-head, as is sometimes used. *D* is a small pulley placed near and on a level with the pin in the lever, for the indicator cord to pass over.

While this form of reducing lever is commonly made with a *pin* for attaching the indicator cord, greater constancy of motion for the drum would be attained by the use of a sector, such as is shown at S in Fig. 15.

To find the point on the lever at which to attach the indicator cord, proceed as follows: Divide the length of the lever by the length of the piston stroke, and multiply the quotient by the required length of the diagram, all expressed *in inches and decimals* of an inch, and the product will be the proper distance from the pivot in the top of lever to the point of attachment.

For example: If the lever is 48 inches long and the piston stroke is 30 inches, and we wish to obtain a diagram 3½ inches long, we have $48 \div 30 = 1.6$; $1.6 \times 3.5'' = 5.6''$, the radius required to give a 3½ inch diagram. If we require a diagram 4½ inches long, then: $1.6 \times 4.5'' = 7.2''$, the radius required to give a 4½ inch diagram.

The object of all mechanisms for actuating the drum of the indicator should be such that the relation of piston to drum movement will be constant. Such constancy cannot, however, be fully attained by the use of any form of reducing lever, and so should not be employed when important adjustments or tests are to be made. Their simplicity and the small expenditure of time and money in their construction may entitle them to favorable consideration on the part of beginners in the use of the indicator, and when only ordinary work is to be done.

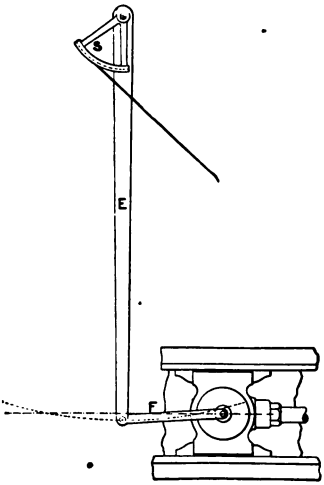


FIG. 15

The forms shown in Fig. 16, Fig. 17, Fig. 18, and Fig. 19 are correct in principle, and when carefully constructed may be relied upon to give correct results.

The Brumbo pulley, shown in Fig. 15, is another form of reducing lever, and one often used by engineers, especially on locomotives. It can be quickly and cheaply made and can be used on almost any kind of engine. The swinging lever E is a strip of straight grained pine, one inch or more in thickness, three to four inches wide, and from one and a half to two times as long as the piston stroke. It is suspended by a bolt or screw, serving as a pivot, from a frame or truss overhead constructed for that purpose, and is connected at its lower end by the wooden link F, to the usual stud or pin fixed in the center or other convenient part of the cross-head; the link should be from one-third to one-half the length of the piston stroke. In the illustration the proportions of lever and link are as 60 to 15; the lever being two times and the link one-half the length of stroke.

The sector S may be constructed of wood or of metal, as here shown; it has a groove in its circular edge for the cord to run in and is screwed to the upper end of the lever or pendulum, so that its center will coincide with the center of the pivot on which it swings. The radius of the sector which is necessary to give the proper motion to the drum to obtain the desired length of diagram can be found as follows: Divide the length of the lever by the length of the piston stroke and multiply the quotient by the length of diagram desired, and the product will be the required radius, all the terms being expressed in inches and decimals of an inch. For example: If the lever is 30 inches long and the piston stroke is 20 inches, and we wish to obtain a diagram 3 inches long, we have $30 \div 20 = 1\frac{1}{2}$; $1\frac{1}{2} \times 3'' = 4\frac{1}{2}''$, the radius required to give a 3 inch diagram.

When the conditions are favorable, the lever should be hung so that it will swing in a vertical plane, parallel with the guides and in line with the indicator, as this arrangement is the most simple, and the use of guide pulleys is avoided. It is not absolutely necessary, however, that the lever shall swing in a vertical plane, but it may swing in a plane at any angle thereto, where the conditions require it. In such cases a man's ingenuity and inventive *faculty must aid him.* A link made of a thin strip of steel, that will twist a little, is in some cases found very convenient.

When the cross-head is at midstroke the lever must hang plumb and the pin which connects its lower end to the link must be as much *below* the horizontal line of motion of the *stud* in the cross-head as it sweeps *above* that line at either end of the stroke. See the dotted line for an illustration of this point, which is important. The cord must lead from the sector in about the same plane with its swing.

Carrying pulleys should be avoided as far as possible, but whatever number is necessary should be firmly placed. The swinging arm of the guide-pulley on the indicator should always be fixed in the direction from which the cord is received.

A piece of piano wire is often used to replace the string leading from the sector to the indicator.

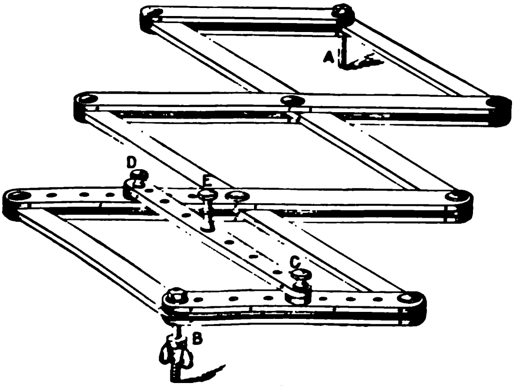


FIG. 16

The pantograph, illustrated in Fig. 16, is another style of reducing motion. Although theoretically it gives a perfect motion, owing to its many joints it may become shaky and give erroneous results, unless it is very nicely made and *carefully used.* When the indicator is applied to the side of the cylinder the pantograph works in a horizontal plane. The pivot end B rests on a post or other support set opposite to the middle of the guides, and the working end A receives motion from the cross-head — to which it is attached by a suitable iron with a hole drilled in it for the stud A to work in. By adjusting the support for the pivot end to the proper height and at a proper distance from the guides, the cord may be carried directly from the pin E to the indicator without the need of carrying pulleys. The movable bar may be set forward or backward by the pins C, D, so as to perfectly adjust the movement of the pin E to the required length of the diagram; this pin must always be in a straight line with the stud A and the pivot B. The string from E should always lead off in a line parallel to the piston rod.

FIG. 17

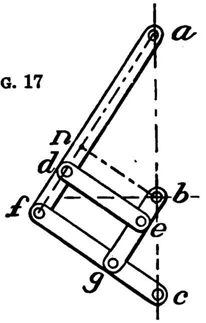


FIG. 18

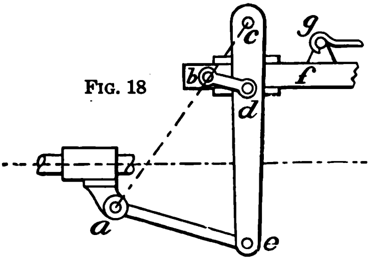
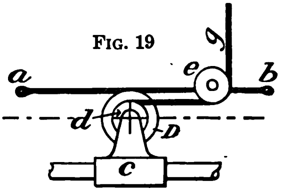


FIG. 19



The directions here given for constructing and arranging drum motions are general; special cases may require modification of the forms and special adaptation of the means here described, all of which call forth the ingenuity and skill of the engineer.

Fig. 17 shows a pantograph device at midstroke. This is made of bar iron. The pins *d*, *e*, *f*, *g*, are nicely fitted. The indicator cord may be attached at *b*. The end *a* is attached to a pin on the cross-head. The fixed fulcrum is at *c*. *a*, *b*, and *c* must always lie in the same straight line, and *e d*, *b n*, parallel and equal to *f g*. Also, $a f : n f =$ stroke of piston to length of indicator diagram.

Fig. 18 illustrates a device used at the Massachusetts Institute of Technology. *f* is a rod moving in a slide parallel to the piston-rod. The link *b d* is attached to *f*, and the link *a e* to the cross-head. *a*, *b*, and *c* must always lie in the same straight line. $a e : b d$ and $e c : c d =$ stroke of piston to length of indicator diagram. The cord is hooked on a pin at *g*; it is well to have a pin for each indicator used.

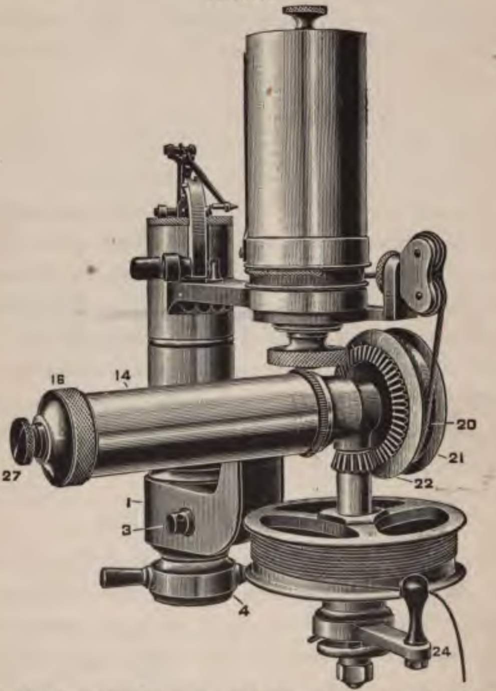
Fig. 19 is a device by Armand Stévart for long strokes. *a* and *b* are fixed ends of cord wrapped around a pulley *D*. The indicator cord is attached to a small pulley *d* and passes around a guide pulley *e*. *D* and *d* are attached to the cross-head. Dia. $D \div$ dia. $d =$ stroke of the piston $\div$ by the difference between stroke of piston and length of card.

The reducing wheel is another device for giving the proper motion to the paper drum. Although old in principle, and as formerly made not highly approved by careful engineers, it is now coming into more general use, and the superior manner in which it is designed and constructed seems to warrant this change.

CROSBY REDUCING WHEEL

The Crosby reducing wheel is attached directly to the cylinder cock of the steam engine, and has connected to it the steam engine indicator which it is to serve; thus it forms a base or support for the latter, and receives all the strains and shocks in the operations of the engine, to *the relief* of the indicator. All its parts are designed and constructed for strength, accuracy, and durability. Its bearings are not only nicely adjusted, but are made comparatively frictionless by the introduction of balls running in hardened tool steel bearings, affording lightness and freedom of movement. It has a helical spring which is more active in recoil than the volute spring in common use, this being a very essential feature for accurate results on high speed engines. The cord pulley is horizontal to allow the cord leading to the engine cross-head to take any direction the circumstances may require without regard to the position of the indicator.

Patented



This Cut Shows the Crosby Standard Steam Engine Indicator *Mounted upon the* Crosby Reducing Wheel

Special tools have been provided for making it, so that like parts are interchangeable, and when worn or destroyed others can be easily substituted; in other words, everything has been done so far as possible to make the instrument in all respects as excellent for its purpose as is the Crosby indicator.

It is adapted to receive any steam engine indicator or indicator cock by means of interchangeable bushings; and by a series of speed pulleys it can be adapted to all steam engines having strokes between the limits of 14 inches and 72 inches.

Whenever the reducing wheel is to be attached to a vertical engine an elbow nipple is provided, which will allow the cord pulley to travel in the proper plane for guiding it to the cross-head of the engine, with the indicator in an upright position as usual.

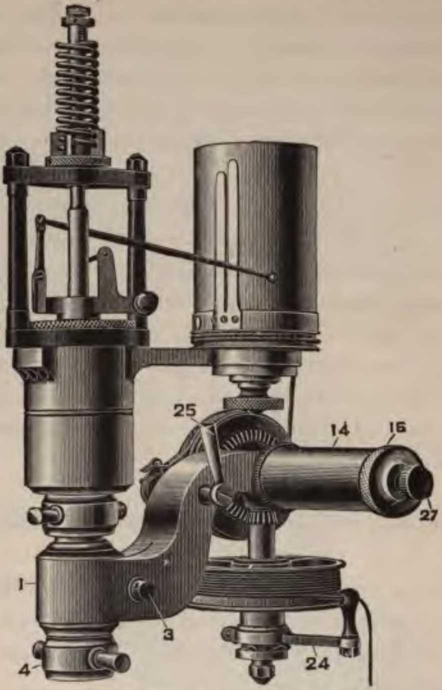
CROSBY REDUCING WHEEL WITH DETENT

Patented

The detent applied to the Crosby Reducing Wheel does not affect the connection between it and the engine, and does not allow the cord leading from the indicator drum to the reducing wheel to slacken.

When the clutch is thrown in to stop the motion, the indicator drum is revolved to the end of the stroke and held there by the drum cord, while the mechanism of the detent controls the cord leading from the reducing wheel to the cross-head of the engine.

When the clutch is released and the motion of the engine is again communicated to the drum, the latter takes up the motion without shock from the point where it stopped, because it starts from a state of rest at the end of the stroke. This is important, for if a drum is stopped and held by a detent in midstroke where the piston is running at its highest speed, at the release of the detent the drum will necessarily start again at such highest speed with a shock.



THIS CUT SHOWS THE CROSBY NEW INDICATOR MOUNTED UPON THE CROSBY REDUCING WHEEL WITH DETENT

Moreover, as such a detent must engage at the highest speed, it often fails to operate and always wears rapidly.

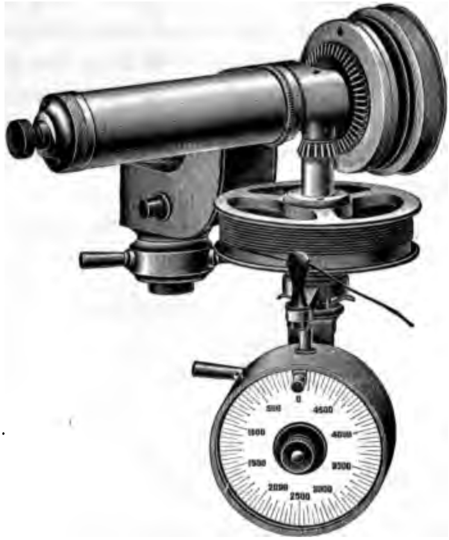
Directions for using the Crosby Reducing Wheel, either without or with detent, are sent with each instrument.

CROSBY REDUCING WHEEL WITH RECORDING COUNTER

Patented

To determine the number of revolutions of the engine per minute an ingenious and convenient device is shown by the cut, representing the Crosby Reducing Wheel having attached to it the Crosby Recording Indicator Counter.

The latter is actuated by the moving parts of the reducing wheel and records on a chart every revolution of the engine; so that during the taking of the diagrams by the indicator attached to the reducing wheel the revolutions of the engine are recorded simultaneously.



Its capacity to record 5,000 revolutions permits its use during a considerable period of indicating work; and the average number per minute so determined is more accurate for such purpose than if the revolutions were merely counted intermittently by the ordinary speed instruments. Besides, there is thus preserved by the chart a record of the work done, to be filed with the diagrams taken by the indicator for future consideration.

It has recorded upwards of 4,000 revolutions per minute without a fault. No difficulty will arise in its attachment and use.

After the reducing wheel has been adjusted to the stroke of the engine, the counter is attached according to the following directions :

Loosen the clamping nut on the back of the counter ; raise the lever to its vertical position, and if the operating pin does not drop, press it lightly downward. Loosen the hexagon nut below the guide bracket of the reducing wheel and slip the fork of the counter bracket under the nut; adjust the height of the counter on its bracket so that the operating pin of the counter when down will just clear the guide bracket; tighten all the nuts securely. To start the counter, throw the finger lever down ; to stop it, raise the lever to its vertical position.

Set the chart at zero. Note the time of starting and stopping on the face of the chart, where indicated.

IMPORTANT SUGGESTIONS

In all cases the indicator cord should be of the right length to prevent the paper drum from recoiling against its stop; and before attaching it to the cross-head of the engine it should be drawn out its full length to ascertain whether or not the cords on the indicator and reducing wheel have been properly adjusted.

All the working parts must be kept well oiled.

The reducing wheel is adapted to be used with a steam engine indicator having either a 1½ inch or 2 inch drum. As indicators with 2 inch drums are now more commonly used, the reducing wheel is ordinarily provided only with stroke pulleys for such size drum. If the reducing wheel is to be used with an indicator having the 1½ inch drum, it should be so stated in order to receive the stroke pulleys of the proper size.

These stroke pulleys are provided in sets and can be so obtained. With the 2 inch drum, the proper stroke pulleys will give cards 4 inches long, and with the 1½ inch drum *the stroke pulleys are* calculated to give cards 3 inches long.

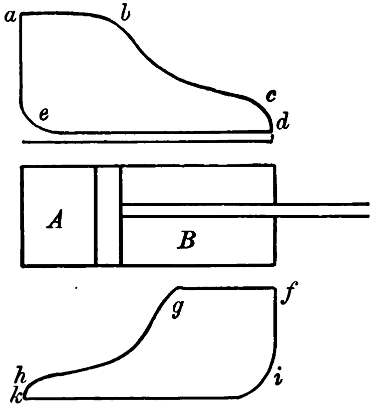
The Mean Effective Pressure, M. E. P. (improperly named perhaps) used in figuring horsepower is calculated from the indicator cards as already explained.

The effective pushing pressure or pulling pressure per square inch of piston area at any point of the stroke is evidently the difference between the total pressures on the two sides of the piston at that point divided by the piston area. Obviously the average pressure, calculated in this way, would not be the same as the mean effective pressure.

(See discussion of stroke cards given later on.)

That the horsepower obtained is correct when the M. E. P. is calculated as previously explained is shown by the illustration which follows.

Let the mean pressure above the atmospheric line corresponding to line $a\ b\ c\ d = P_f$.



Let the mean pressure above atmospheric line corresponding to line $f\ g\ h\ k = P_b$.

Let the mean pressure above the atmospheric line corresponding to $k\ i\ f = E_b$.

the card, and cards should be taken with as faint lines as possible on this account.

Make notes on the card of as many of the following facts as possible. The day and hour of taking the diagram; the kind of engine from which the diagram is taken, and which engine, if one of a pair; which end of the cylinder, the diameter of the cylinder, the length of the stroke, the diameter of the piston-rod, and the number of revolutions per minute; the position of the throttle; the atmospheric pressure; the steam pressure at the boiler and at the engine, by the gages; the vacuum by the gage on the condenser and the temperature of the feed at the boiler; if the engine is compound, the pressure in the receiver; the scale of the spring used in the indicator; the volume of the clearance at each end of the cylinder, and what per cent of the piston displacement each of these volumes is. (Directions for ascertaining the volume of the clearance, and what per cent that volume is of the piston displacement, are given on pages 94 and 96.)

It is often useful to make notes of special circumstances of importance, such as a description of the boiler, the diameter and length of the steam and exhaust pipes, the temperature of the feed water, the quantity of water consumed per hour, etc.

On a locomotive, note the time of passage between mileposts in minutes and seconds, from which, when the diameter of the drivers is known, the number of revolutions per minute may be calculated. Note also the position of the throttle and the link, the size of the blast orifice, the weight of the train, and the gradient.

On diagrams from marine engines note, in addition to the general facts, the speed of the ship in knots per hour, the direction and force of the wind, the direction and state of the sea, the diameter and pitch of the screw, the kind of *coal, the amount* consumed, and the ashes made per hour.

CHAPTER IX

HOW TO FIND THE POWER OF AN ENGINE

To find the power actually exerted within the cylinder of a steam engine, it is necessary to ascertain separately *three factors* and the product of their continued multiplication. These factors are: The area of the two sides of the piston; the total travel of the piston in feet per minute; and the mean effective pressure urging the piston forward, designated M. E. P.

The piston area. This, at the back end, is the same as the area of cross-section of the cylinder; at the crank end it is the same, less the area of cross-section of the piston-rod. These areas may be found from their diameters in a table of the areas of circles, or may be computed by multiplying the square of the diameter in inches by the approximate number 0.7854.

The travel of the piston. The total travel of the piston in feet per minute is found by multiplying twice the length of the stroke measured in *feet*, by the number of revolutions of the crank shaft per minute, which should be carefully ascertained by taking the mean of many countings, or the readings of a speed counter during a considerable time. The mean piston speed will be expressed in terms of *feet per minute.*

The mean effective pressure. There are several approximate methods for computing the mean effective pressure, one of which is to divide the diagram into ten equal parts, as shown in Fig. 20. Then through the points of division draw lines, which are called ordinates, at right angles to the atmospheric line. The mean heights or pressures of the small areas thus formed are indicated by the dotted lines midway between the ordinates.

missing has prevented any great amount of heat from being given by the cylinder to the new charge during intake and compression and consequently the lessened pressure at the end of compression causes a greater reduction in the mean effective pressure than is gained by the elimination of the carbonic acid gas from the clearance space.

To get a fair average card the pencil of the indicator should be left on the paper during a complete firing cycle.

The spring used in a gas engine indicator is ordinarily one hundred and fifty or two hundred, and variations of two or three pounds between the atmospheric line and the intake and exhaust lines are hardly noticeable with these springs and have not generally been considered in getting the mean effective pressure of the card. This resistance through the intake and the exhaust valves should be taken into account in getting the power developed by the engine. By neglecting these, errors of from ten to twenty per cent may creep into the work. To study these resistances it is necessary to use a light spring in the indicator. This spring may be protected by a ferrule or stop, furnished with the gas engine indicator, which prevents the piston from moving above a certain level.

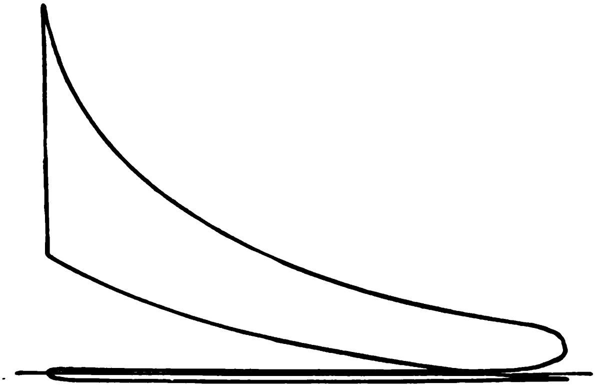


FIG. 23a

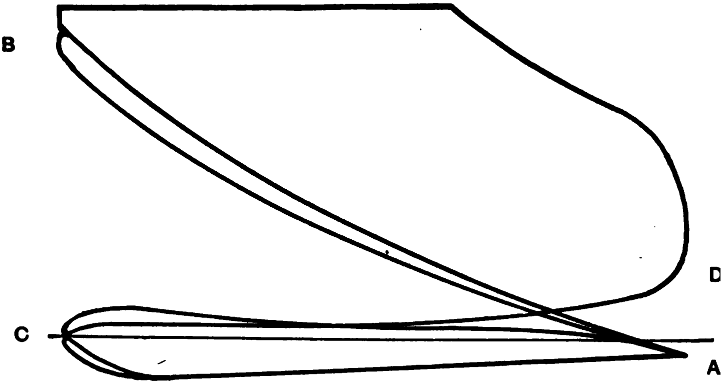


FIG. 23*b*

Cards taken with a stiff spring and with a light spring are shown in Figs. 23*a* and 23*b*. The straight line at the top of Fig. 23*b* is drawn by the pencil while the indicator piston is against the stop. The line D C is the exhaust line and C A is the intake. The distance across this loop will vary from four to eight pounds, according to the make of engine.

The card shows also the compression and expansion of air on a "miss" stroke. The expansion of the air is at a pressure slightly lower than the compression, thus making a negative loop. It will be noticed that the last end of the expelling stroke shows a higher pressure when forcing out air than when forcing out burned gas. The greater velocity of the moving column of burned gas in the exhaust pipe exerts more of an aspirator action at the cylinder at the end of the exhaust stroke and so causes a lower pressure.

To obtain the mean effective pressure for an engine working on the "hit and miss" method of governing: Calculate the mean effective pressure, using compression and expansion lines for a card which represents the mean card of a complete firing cycle of working and missing strokes. Find from the card taken with the light spring the mean pressure corresponding to the loop made between the exhaust and intake lines and also the mean pressure representing the distance between the compression and the expansion lines during a miss period as shown by A B, Fig. 23*b*. This correction is to be applied only in the proportion that the misses stand to the firing charges.

Suppose the M. E. P. as figured from the card taken with stiff spring, neglecting the loops at the bottom of the card, was 89 pounds. Assume the bottom loop to amount to 7 pounds. Assume the air loop to amount to 3 pounds, and call the firing cycle fire 5, miss 2, etc. The correct M. E. P. is $89 - 7 - (\frac{2}{5} \times 3) = 80.8$; if the value 89 had been taken for M. E. P., the error in the power calculation would have been over ten per cent. Suppose the cycle had been fire 2, miss 5, etc., and other values the same as above. The correct M. E. P. $= 89 - 7 - (\frac{5}{2} \times 3) = 74.5$; by neglecting these loops an error of 19 per cent would result.

If the engine fires every fourth stroke and governs by throttling the charge, it is even more important than in the previous case to get the cards with a light spring.

After obtaining the correct M. E. P., the H. P. of a gas engine is figured by multiplying the area of the piston in square inches by the M. E. P. and by the number of feet traveled by the piston per minute *during working strokes*, and dividing by 33,000.

An engine governing by "hit and miss" must have some form of positive counter or indicator attached, so as to give the number of explosions in a certain time.

Engines frequently give trouble by back firing. This is in some cases due to the fact that the mixture is still burning in the cylinder at the time the exhaust valve closes and the new charge enters; in other cases a long cored hole, a *port passage or* an indicator pipe may hold a burning mixture all through the exhaust stroke and up to the time the fresh charge enters. This is more apt to be true with slow burning mixtures, which are weak, than with rich mixtures. In some instances the indicator piping has had to be made of smaller size and an indicator with a very small piston used, in order to prevent this firing of the incoming charge.

The theoretical efficiency of a four-cycle gas engine is

$$\text{Eff.} = 1 - \left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}}$$

where p_1 is the absolute pressure at the beginning of compression, p_2 is the absolute pressure at the end of compression and k is the ratio of the specific heats of the unburned mixture at constant pressure and at constant volume.

Two-cycle engines. An indicator with light spring should be attached to the crank case or compression chamber of the small two-cycle engines in addition to that on the working cylinder. The work of compression in the crank case is to be deducted from that shown by the working cylinder. Large two-cycle engines compress the gas and air in separate compressors.

ACTUAL THERMAL EFFICIENCY OF A GAS ENGINE

Tests made on large gas engines show that of the heat of combustion of the gas supplied, about forty per cent is carried off in the jacket water, about thirty-five per cent goes off in the exhaust, and only about twenty-five per cent is converted into work, as shown by the card.

Example. A small gas engine uses 22 cubic feet of city gas per I. H. P. per hour. The gas has a heating value of 700 B. t. u. per cubic foot. The heat units supplied per minute are $\frac{22 \times 700}{60} = 257$; a horsepower is 33,000 foot-pounds per minute, equivalent to $\frac{33{,}000}{778} = 42.42$ B. t. u. The thermal efficiency per I. H. P. is $\frac{42.42}{257} = 0.165$, or 16.5 per cent.

CHAPTER X

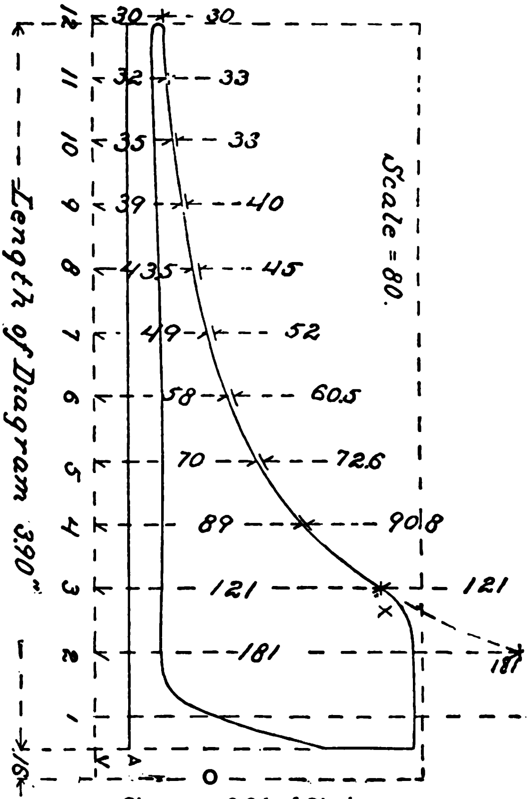
THE HYPERBOLIC CURVE

This curve is frequently applied to indicator diagrams for the purpose of comparing it with the expansion curve, as drawn by the indicator, and if it coincides pretty nearly, this fact may generally be taken as evidence tending to show that the steam and exhaust valves of the engine are properly closed and the piston tight.

Without going into any discussion regarding condensation and re-evaporation in steam engine cylinders, it is a well known fact that indicator diagrams, taken from large engines, properly made and in good order, show expansion curves which are close approximations to the hyperbola.

Before this curve can be drawn, it is necessary to ascertain the capacity of the clearance or waste room: that is, all the space between the cylinder heads and the piston at each dead-center, including the counterbore and the ports up to the face of the closed valves.

There are several ways of finding this: one, by direct calculation from sectional drawings, when accurate drawings can be obtained; another, by putting the engine at dead-center with valves closed, and then filling the clearance space with water, which has been carefully weighed in a convenient vessel, then weighing what is left. The difference between the weight of the whole and the remainder is the weight of water required to fill the clearance space. From this the number of cubic inches occupied by the water may be computed. At ordinary temperatures (60° to 75° F.), for all practical purposes, we may call the weight of one cubic inch of water 0.036 pounds and 27.8 cubic *inches of water* equal to one pound. Then the number of pounds of water divided by 0.036 or multiplied by 27.8 will give the number of cubic inches. If accurate scales for weighing the water are not at hand, it can be carefully



Clearance 0.04 of Stroke

FIG. 24

measured in a quart or pint measure and the number of cubic inches found directly. A gallon contains 231 cubic inches, a quart 57.75, and a pint 28.875 cubic inches.

The volume of the clearance will rarely be the same at the two ends of the cylinder, therefore the number of cubic inches in the clearance at each end must be divided by the net area of the piston at its own end: that is, the number of cubic inches in the clearance at the end nearest the crank must be divided by the number of square inches in the cross-section of the cylinder, less the number of square inches in the cross-section of the piston-rod; and the number of cubic inches in the clearance at the end farthest from the crank must be divided by the number of square inches in the cross-section of the cylinder. The quotient in each case will be the length of clearance at the respective ends of the cylinder, expressed in inches. In this instance (Fig. 24) it is found to be 0.16 of an inch.

It is convenient to have the length of the clearance expressed as a fraction of the piston displacement or stroke of the piston. To obtain this fraction, divide the number of cubic inches in volume of clearance by the number of cubic inches in the volume swept through by the piston at each end separately, taking care to allow for the volume occupied at one end by the piston-rod, and the quotient will be the decimal fraction that the clearance space is of the volume swept through by the piston.

Fig. 24 illustrates a good method for locating points in the hyperbola through which the curve may be drawn.

First, draw the zero line V, at the proper distance, viz., $14\frac{7}{10}$ pounds by the scale, below and parallel with the atmospheric line; next, draw the clearance line O, as computed, 0.16 of an inch from the end of the diagram; next, locate the point of cut-off X, and draw the perpendicular line numbered 3 through it; next, divide the space between this line and the clearance line into three equal parts; then, taking one of these parts for a measure, point off, on the vacuum line, *equal* spaces toward the left hand until one or more falls beyond the end of the diagram as shown, and erect perpendicular lines from each point. These lines are *called ordinates* and numbered consecutively 1, 2, 3, 4, etc., beginning with the one next to the clearance line. It is well to bear in mind the fact that vertical distance on a diagram represents *pressure* and horizontal distance *volume*.

In this case we have started the hyperbola from the point of cut-off X, and its course is indicated by the short lines drawn through the ordinates a little above the actual curve, with their calculated pressures written above; the actual pressures of the expansion curve are written below it. The properties of the hyperbola are such, that if the distance of the point X from the clearance line O be multiplied by the height of X from the zero line V, the height of any other point in the curve can be found by dividing this product by its distance from the clearance line. And on this principle we proceed to locate points on the ordinates through which our hyperbola will run.

We find the pressure at the point of cut-off to be 121 pounds with a volume which we call 3, because there are three spaces or volumes between it and the clearance line. Then, $121 \times 3 = 363$, which is our dividend for all the other volumes. Therefore, the height at which the hyperbola will cut ordinate 4 will be determined by dividing 363 by 4, which is 90.8 (it is unnecessary to carry the division beyond one decimal); and at ordinate 5, 72.6; at ordinate 6, 60.5; and so on to the end. At ordinate 12 we find that the hyperbolic and the actual curves practically coincide. In like manner we may extend the curve to the right: $363 \div 2 = 181$ pounds, which would be the pressure if the steam were compressed up to 2 volumes. If desired, the hyperbolic curve can be started just before the point of release and projected in the opposite direction by the same method.

Instead of using figures which stand for pressures or volumes of steam to locate the hyperbola, as in this instance, the distances from the base and perpendicular lines of any point may be expressed in inches and decimal parts, with the same result.

A quick way to draw the hyperbola is to take the whole distance between ordinate 3 and the clearance line, (Fig. 24), as a measure, and set off equal spaces to the left as before directed. Then we would have but four ordinates and would number them as follows: 1 at 3d, 2 at 6th, 3 at 9th,

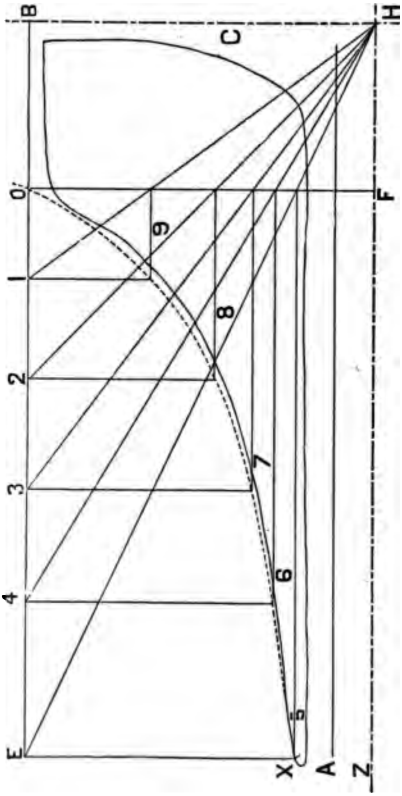


FIG. 25

and 4 at 12th. At 1 we would have a pressure of 121 pounds; at 2, 121 pounds $\div 2 = 60.5$; at 3, 121 pounds $\div 3 = 40.3$; *and* at 4, 121 pounds $\div 4 = 30.25$.

As a general rule, the near approximation of the expansion curve to the theoretical or hyperbolic curve may be taken as evidence of good conditions but should not be accepted for a certainty, unless all the known facts and conditions tend in the same direction.

GEOMETRIC METHOD OF FINDING THE HYPERBOLA

The hyperbola may be found by following the directions given below, in connection with Fig. 25. A is the atmospheric line; Z the zero line, or line of no pressure; B the line of boiler pressure, and C the clearance line. Locate the first point in the hyperbola at the point of release X and draw the vertical line X E. Then draw diagonal line, E H; then, from X, draw horizontal line 5 to its intersection with E H, through which draw vertical line F O. Now mark off points between O and E as 1, 2, 3, 4 — exact spacing is unnecessary — and from these points draw diagonal lines to H and vertical lines down to, or a little below, the actual curve. Now draw horizontal lines 6, 7, 8, and 9 from the points of intersection in the line F O, of the diagonal lines H 4, H 3, H 2, and H 1 respectively; and the points where these lines cross the vertical lines, 1, 2, 3, and 4, in connection with points X and O, are the points through which the hyperbola should be drawn, as shown by the dotted curve.

ANOTHER METHOD OF FINDING THE HYPERBOLA

Through the point of release *b*, draw any line as *a* B and make A B equal to *a b*, as shown in Fig. 26. Then draw any other line as *c* D and make *c d* equal to A D; then *d* will be a point in the hyperbola passing from *b* to A, as shown by the dotted curve. By drawing a number of lines through A and following the same method, we can find as many points in the hyperbola.

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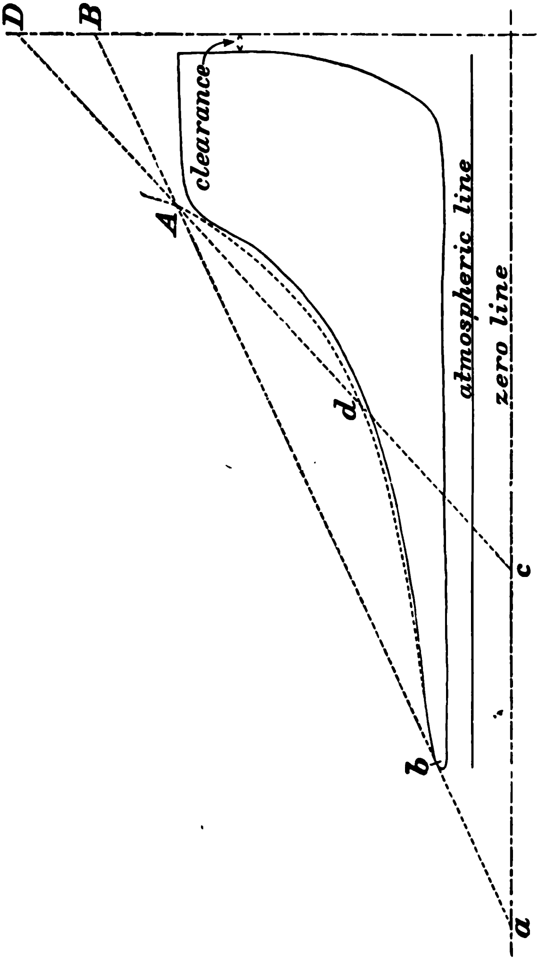


FIG. 26

PART II

PART II

APPLICATIONS

The steam engine indicator may be used for a great many purposes besides those mentioned in the preceding chapters. The applications are so varied that the cases selected and illustrated in these pages show only a few of those which seem to be most useful.

I. Valve Setting

If the cards taken from the two ends of the cylinder resemble Figs. 27 or 28, the eccentric has too little angular advance and must be turned ahead; in the case of an engine giving cards shown by Fig. 27, moving the eccentric ahead

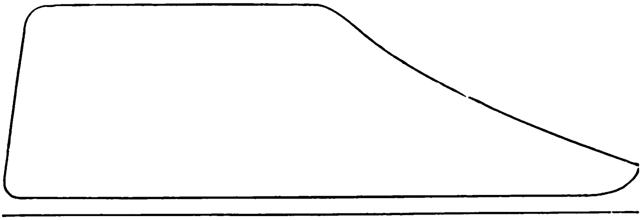


FIG. 27

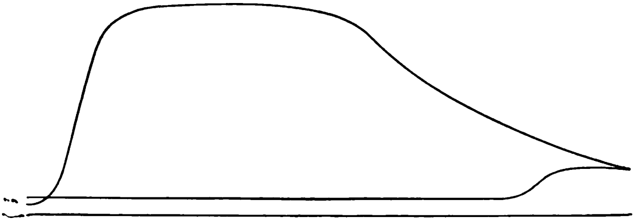


FIG. 28

5° to 10° would probably be sufficient, while with Fig. 28, 20° to 30° would be needed.

If the cards from both ends resemble Figs. 29 or 30, the eccentric has too much angular advance and should be turned back a small amount for Fig. 29, and a much larger amount for Fig. 30.

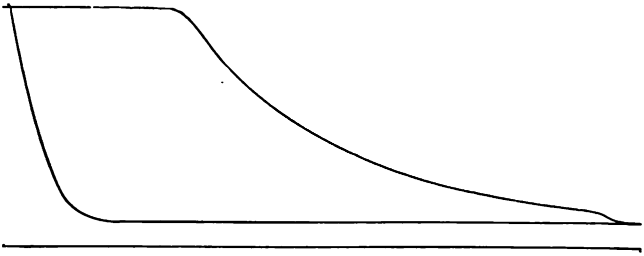


FIG. 29

Cards similar to Figs. 29 and 30 are obtained at early cut-off from most single valve high speed engines with fly-wheel governors.

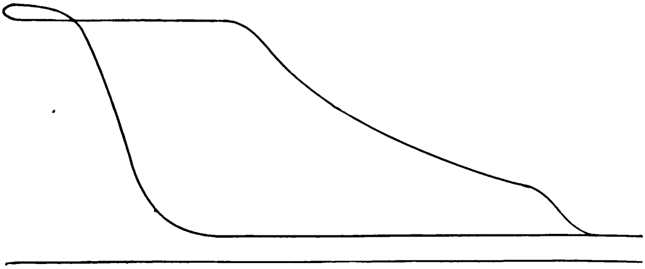


FIG. 30

A plain slide valve engine, or an engine with piston valve, taking steam on the outside, will give cards like Figs. 31 and 32 when the valve spindle is too long, the eccentric being in the right place.

The effects of lengthening the valve spindle are (1) to increase the steam lap on the head end, delaying admission and hastening cut-off on this end; (2) to decrease the steam lap on the crank end, hastening admission and delaying cut-off; (3) to decrease the exhaust lap on the head end, hastening release and delaying compression; and (4) to increase the exhaust lap on the crank end, delaying release and hastening compression.

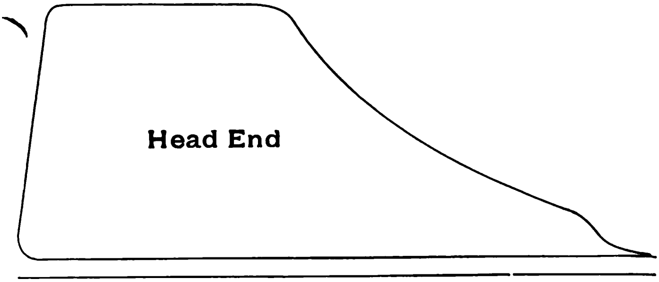


FIG. 31

If the valve spindle is too long and at the same time the eccentric has too little angular advance, or is set too far back, the cards from the engine will be similar to Figs. 33 and 34.

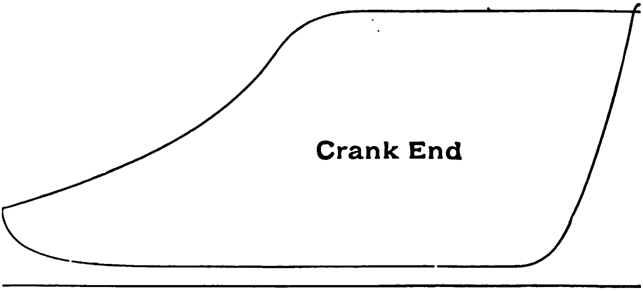


FIG. 32

If the eccentric is set too far ahead and the spindle is too long, the appearance of the cards may be predicted by

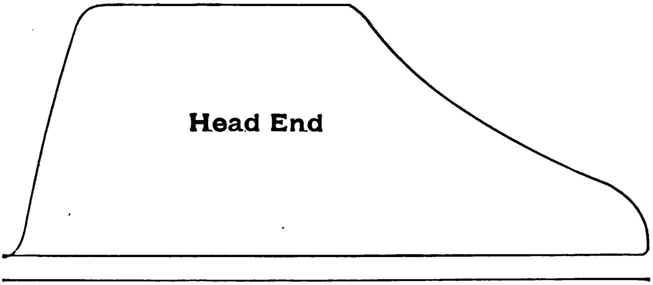


FIG. 33

combining the effects shown by Figs. 29 or 30 with Figs. 31 and 32.

2. Steam Chest Cards

A steam engine indicator may be connected to the steam chest of the engine and cards taken simultaneously with

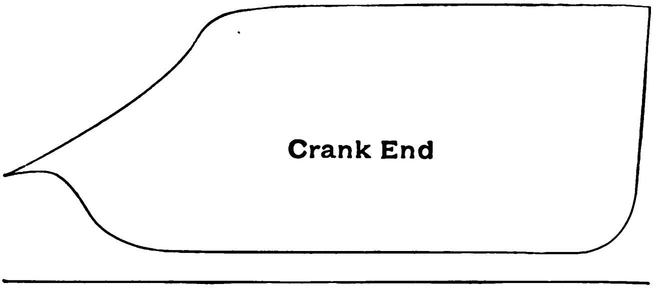


FIG. 34

cards from the cylinder, both indicators being attached to the same reducing motion.

For comparison, it is convenient to superimpose the chest card on the cylinder card, as has been done in Fig. 35.

The steam chest card A B C D E C A, Fig. 35, is for one revolution. If the steam pipe and steam port leading from *the steam chest to* the cylinder are large enough, there will be no appreciable drop in pressure between the boiler and the steam chest, or between the chest and the cylinder. If, however, a chest card similar to that shown by the full line is obtained, the steam pipe is too small. This is shown by the fact that the drop from boiler pressure A appears in the steam chest card while the drop between the steam chest and the cylinder is slight.

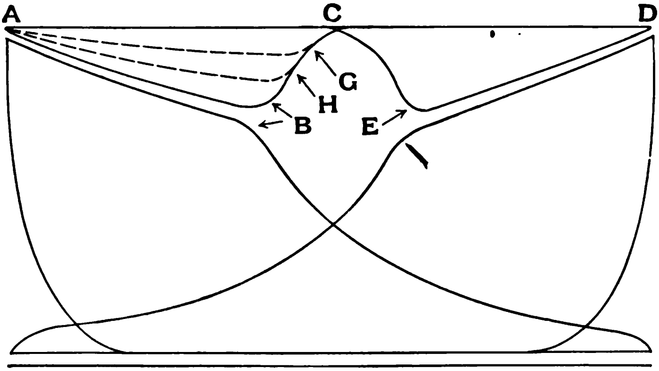


FIG. 35

If the chest card showed a slight drop in pressure, as indicated by the dotted line AG, it would mean that the steam port leading from the chest to the cylinder was too small.

A chest card shown by the line AH would mean that both the steam pipe and the steam ports were too small.

Sometimes the pressure at C is greater than boiler pressure, A. This results from the sudden checking of the velocity of the steam in the pipe supplying the engine.

3. The Eccentric Card

Near the ends of the stroke of an engine the crank turns through a considerable number of degrees without giving much motion to the cross-head.

On account of this fact the indicator card taken in the ordinary way is of little value in investigating any peculiarities which may be noticed at or near the ends of the stroke.

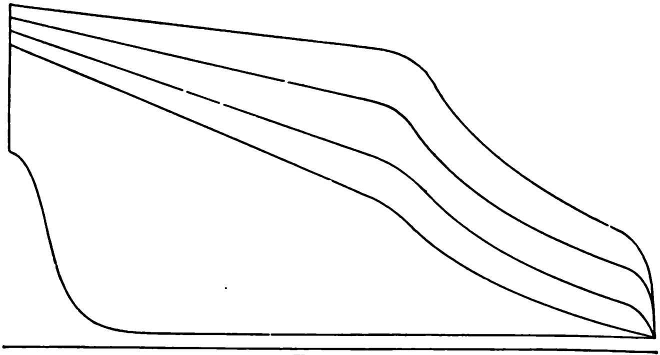


FIG. 36

If, however, the drum motion be taken from the eccentric, which is ordinarily a little over 90° ahead of the crank, the compression and admission lines and the line at release will be spread out at the center of the diagram, while the expansion and exhaust lines will be shortened and appear at the ends of the cards.

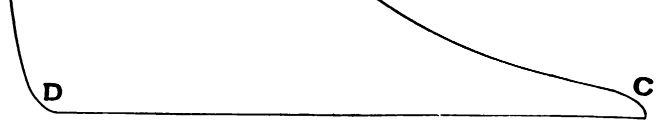


FIG. 37

Fig. 37 represents a good steam card, and Fig. 38 an *eccentric* card.

The cards are lettered at admission, cut-off, release, and compression with the letters A, B, C, D, respectively.

It will be noticed on the eccentric card that the line DA is a reverse curve.

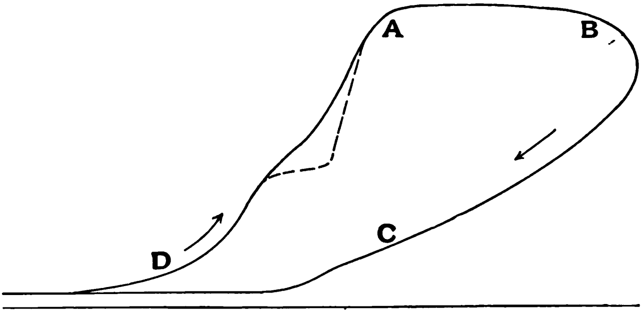


FIG. 38

On the card taken from an engine with throttling governor, Fig. 36, a peculiar drop near the end of compression is found. This is due partly to a leakage of steam from the clearance space over the bridge into the exhaust and partly to a slight amount of condensation. This drop appears on the eccentric card as in the dotted line.

4. Steam Cards from a Westinghouse Air Brake Pump, or from a Wet Air Pump with Surface Condenser

In the diagram shown by Fig. 39 the piston starts at the left and moves towards the right. As the work opposed to the steam piston by the air piston is but trifling during the first part of the stroke, the steam piston "runs away from the steam," causing the drop in pressure shown.

Near the end of the stroke the extra work opposed to the steam piston by the air piston causes it to slow down, and then steam has time to fill the cylinder at nearly boiler pressure. On the return stroke the piston, during the first part of its stroke, travels so fast that the steam cannot be freely exhausted. During the latter part of the stroke, however, when the piston slows up, the pressure drops nearly to that in the exhaust pipe, as shown by the lower line of Fig. 39.

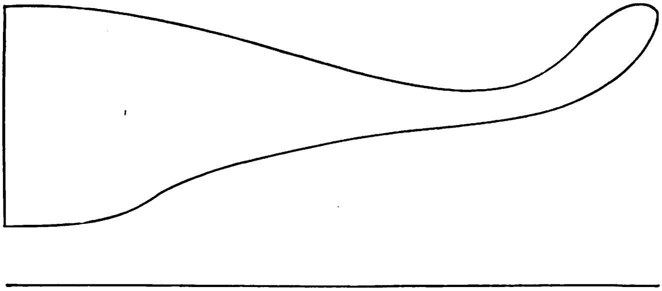


FIG. 39

A casual inspection of Fig. 39 might lead to the conclusion that the greatest effective pressure on the piston was at the left-hand end near the beginning of the stroke. This is not the case, as may be seen by constructing what is called the stroke card, shown by Fig. 40, and again by Fig. 46. Assuming that the piston has a tail-rod so that the area of the head end of the piston is the same as the area of the crank end, the effective push per square inch on the piston at any time is the difference of pressure on the two sides at that time. If now the steam line of the card from

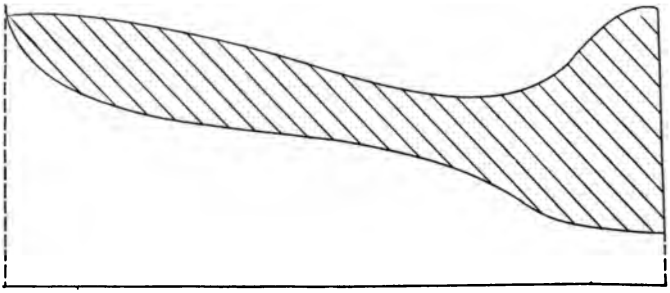


FIG. 40

the head end be combined with the back pressure line from the card of the other end, we get a diagram known as the Stroke Card, sometimes called the True Card, which, by its distance between lines, gives the effective pressure per square inch on the piston at any point. Such a diagram is shown by Fig. 40 for the Air Brake Card. It will be noted that the greatest effective push on the piston comes at the *right*-hand end, not the left.

If the two ends of the steam piston have not the same area, the pressures from one indicator card may be multiplied by the ratio of the two areas, before plotting, in order to reduce to the same basis for comparison.

5. Air Compressor Cards

Figs. 41 and 42 are respectively from an air compressor and from the air end of a certain type of air pump.

The irregularities in the delivery line of Fig. 41 are due to the vibrations of the delivery valve. The slight drop at the beginning of air intake results from resistance to opening offered by the suction valves.

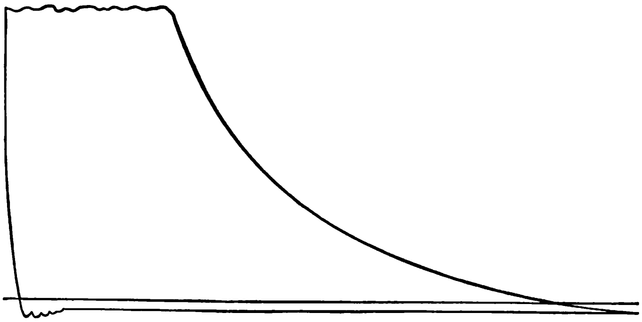


FIG. 41

The air pump pumps a mixture of water and air. On account of the large clearance the curves are much less steep than in the case of the air compressors.

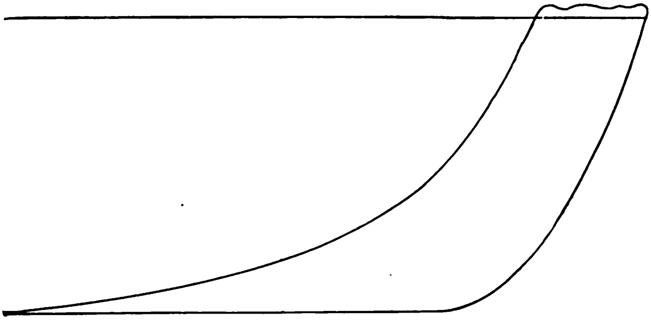


FIG. 42

6. Gas Engine Cards

Figs. 43 and 44 are from an Otto Gas Engine. Fig. 44 has the firing delayed till the end of the stroke, and shows plainly the effect of not having a "lead" on the firing spark.

Engines working on the Otto cycle can have only one working stroke in four, and as many of these engines work on the "hit and miss" principle, in figuring the horsepower from the cards it is necessary to note the actual number of explosions per minute.

Starting at exhaust on the right-hand end, the piston

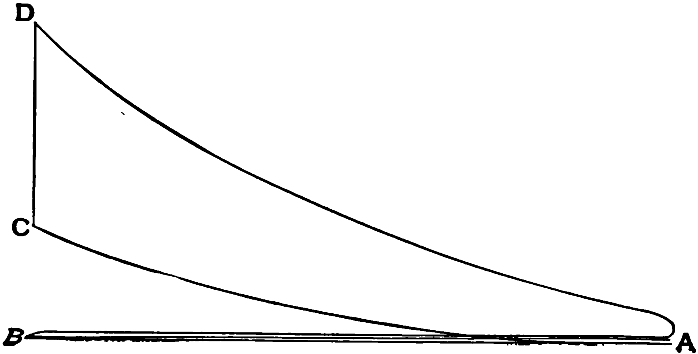


FIG. 43

moves to the left, along AB, and drives out the burned gases through the exhaust valve; next the exhaust valve closes, and as the piston moves to the right, gas and air are drawn in, along BA, the mixture being regulated by the opening of the gas inlet, so as to get the proper ratio of gas to air. On the third stroke this mixture is compressed, along AC, and just before the piston reaches the end of the stroke the mixture is fired; the hot gases expand during the fourth stroke, DA.

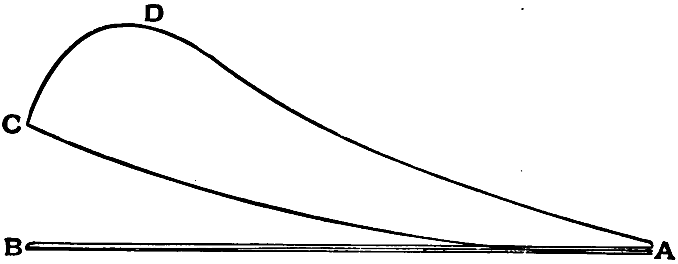


FIG. 44

If the engine is up to speed and gas is not admitted by the governor, the cylinder is filled with air on the filling stroke; this air is compressed on the third stroke and expands back again along nearly the same line on the fourth stroke.

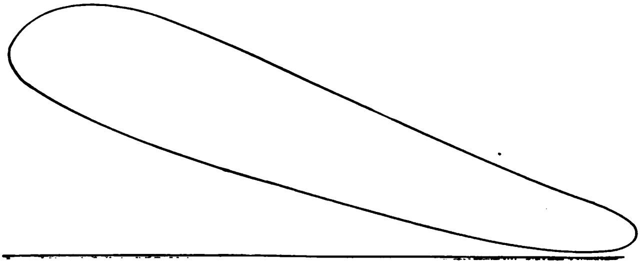


FIG. 45

7. Ericsson Hot Air Engine Cards

Fig. 45 is from an Ericsson hot air engine. This is taken with a 10 pound spring. As there are no valves in this engine, the same air being alternately heated and cooled, it is impossible to get sharp corners on the card.

8. Stroke Card

Fig. 46 represents a stroke card from a Corliss engine. This is constructed as explained in connection with Fig. 40.

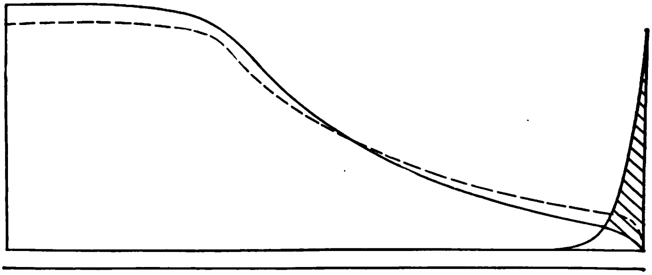


FIG. 46

Near the end of the stroke the pressure on the opposing side of the piston is greater than that on the pushing side, as shown by the part cross-hatched. During this short interval the fly wheel pulls the engine over.

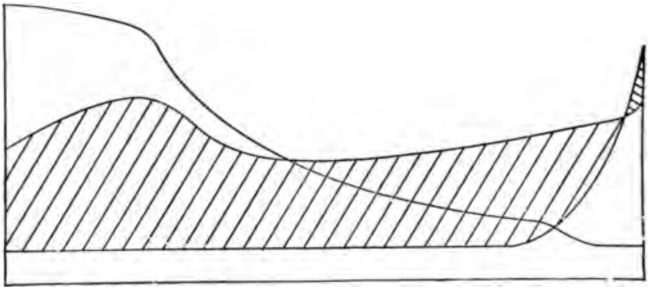


FIG. 47

9. Rotative Effect

The total pressure on the piston-rod at any point may be found by multiplying the pressure measured across the stroke card at that point by the area of the piston. Suppose the total pressure on the piston-rod at the position shown in Fig. 48 is 1,000 pounds. Draw the line P to represent this pressure at some assumed scale, say, for example, one inch representing 500 pounds. Then P = 2 inches long.

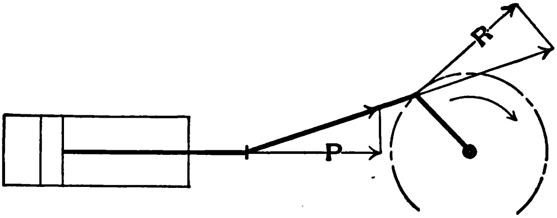


FIG. 48

This force P produces a push along the connecting rod and a downward thrust on the guides. This push along the rod and this thrust on the guides may be found by making a triangle of forces as shown. The length of the different sides of the triangle multiplied by 500 gives these forces in pounds.

The thrust in the connecting rod is seen to be greater than P. This thrust at the cross-head end of the connecting rod is carried to the crank pin where it may be separated into two forces, one force R at right angles to the crank tending to produce rotation, and another force along the crank making a compression in the crank casting.

If, now, values of R are calculated for a sufficient number of points and plotted on a line which represents the development of the crank pin circle, a diagram of rotative effect like Fig. 49 is obtained. The upper half comes from one stroke card and the other half from the other stroke card.

The mean rotative effect is shown by the dot and dash lines.

The amount of energy stored and restored by the fly wheel during a stroke is represented by the area within the curve outside of the dash and dot line.

In this discussion it has been assumed that the entire pressure on the piston was available for producing rotative effect. This is not the case, however. A certain amount of this pressure is used up at the beginning of the stroke in accelerating the piston, piston-rod, cross-head, and a part of the connecting rod. As these are brought back to rest at the end of the stroke, as much energy is recovered as was lost at the beginning. The dotted line on Fig. 46 shows how the upper line should be changed to make allowance for this, in the case of a low speed Corliss engine. On a high speed engine this is a much larger factor, as may be seen in Fig. 47. In this figure the effective pressure per square inch available for producing rotative effect is shown *by the area cross-hatched.*

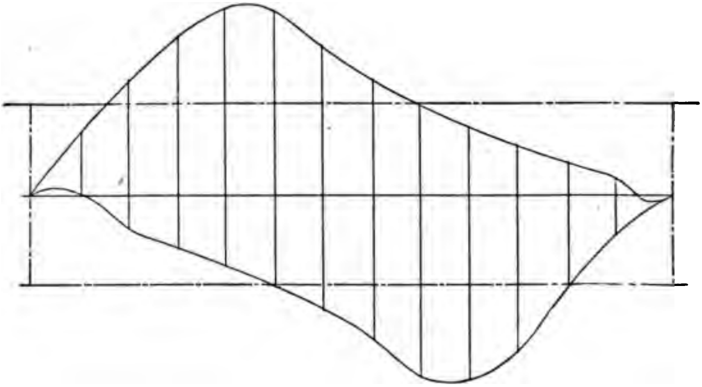


FIG. 49

10. Explosive Force of a Mixture of Gas and Air

Fig. 50 was obtained with a gas engine indicator and a tuning fork. The mixture, which was one part Boston gas to six parts air, was compressed to 25 pounds, as shown at the left of the diagram. It was then fired by a spark, and a pressure of 225 pounds above the atmosphere resulted. The tuning fork, which was marking on the paper at the same time, gave a means of estimating the time required to reach maximum pressure. Each wave denotes $\frac{1}{60}$ of a second.

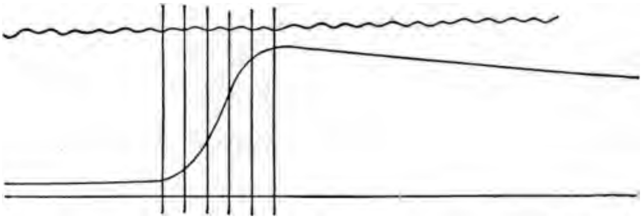


FIG. 50

This explosion was $\frac{5}{60}$ or $\frac{1}{12}$ of a second in reaching maximum pressure.

11. Measuring Water Hammer

Fig. 51 was taken from a 4 inch drive pipe 90 feet long, supplying a Rife Hydraulic Ram. The fall of water to

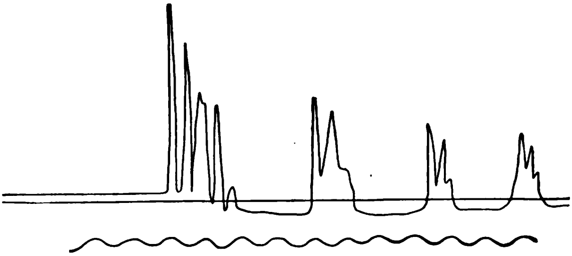


FIG. 51

the ram was about 8 feet. Beneath the card a wave line drawn by a tuning fork gives a means of measuring time accurately. The maximum pressure is 320 pounds. The recurrence of figures similar to the first, but with rapidly decreasing pressures, seems to indicate a wave traveling back and forth in the pipe. The air which is carried by the water no doubt has something to do with this.

12. Flather Dynamometer

A form of dynamometer built by Mr. John J. Flather makes use of a steam engine indicator for measuring power. Figs. 52 and 53 show the indicator cards taken from a dynamometer attached to a power drill. Fig. 52 was taken with a sharp drill running into a cored hole in a casting. The hole was coated with rough sand and the drill lost its edge rapidly, when it gave a diagram like Fig. 53.

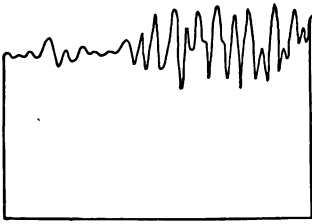


FIG. 52

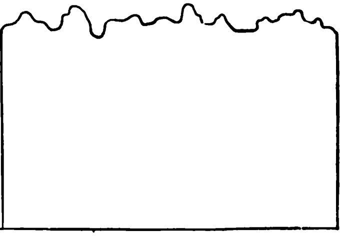


FIG. 53

This dynamometer consists of a hollow shaft on which are two pulleys, one tight and the other loose. On opposite

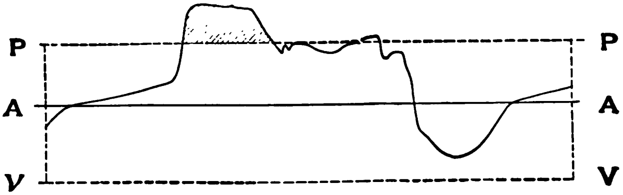


FIG. 54

arms of the tight pulley are two small cylinders which are open at one end. The cylinders connect through pipes, at the closed ends, with the hollow shaft. Built out from the arms of the loose pulley are two studs carrying pistons which fit into the cylinders. The cylinders are filled with oil, as is also the hollow shaft, to the end of which the indicator is attached. The loose pulley drives the tight pulley through its pistons, which press on the oil in the cylinders carried by the tight pulley. The steam engine indicator records this pressure. The drum of the indicator is driven from the shaft through a worm and wheel.

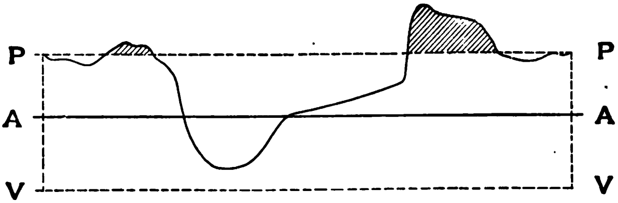


FIG. 55

13. Pulsometer Steam Pump

Figs. 54, 55, 56 are taken from the left side, the right side, and the air chamber of a pulsometer steam pump.

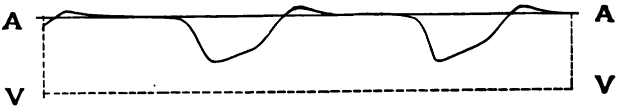


FIG. 56

These cards were taken at the same time. The drums of the indicators were moved at the same uniform rate by means of gears driven by a line of shafting. Fig. 57 shows the combined diagram, giving the pressures at any part of the pump at any time.

The line AA represents the atmospheric line, the line VV that corresponding to an absolute vacuum, and the line

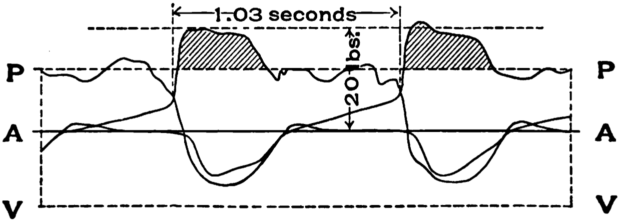


FIG. 57

PP the pressure in pounds due to the hydrostatic head the pump was delivering against. The shaded portions represent deliveries of water.

PART III

PART III

CHAPTER I

PROPERTIES OF STEAM AND OF PERFECT GASES

In order to make complete calculations of an engine test, and in order to get as much information as is possible from the cards, it is necessary to understand the properties of saturated steam and to be able to make intelligent use of tables or plots giving such properties.

It is not the intention to give here any lengthy discussion of thermodynamics, and only such parts of that subject will be touched as bear directly on work depending upon the indicator card or the solution of practical problems such as may come to an engineer in charge of a plant.

Pressures. The pressure of the atmosphere is usually taken as 14.7 pounds per square inch. This corresponds to a corrected reading on the barometer of 29.92 inches. Steam gages are made to read pressures above the atmosphere; therefore to get what is called the *absolute pressure,* the pressure of the atmosphere must be added to that shown by the gage.

A barometric reading in inches of mercury may be reduced to pounds pressure on the square inch by multiplying its corresponding reduced height at 32° F. by 0.491.

The reduced height for a barometer having a brass scale may be calculated from the following:

$$\frac{[1 + 0\ 0000102\ (t - 62)] \times h}{1 + 0.000101\ (t - 32)}$$

where h is the observed height in inches and t is the Fah heit temperature at the barometer.

To measure pressure below the atmosphere, a vacuum gage or a glass U tube filled with mercury may be used. It is customary to quote the vacuum in inches of mercury instead of in pounds. If the barometer stood 29.9 inches a vacuum of 26 inches would mean that there was an absolute pressure of 3.9 inches, or (3.9 × 0.491) pounds.

All pressures as given in plots or in tables of the properties of saturated steam are absolute pressures.

Specific pressure and *specific volume.* Specific pressure is absolute pressure on the square foot. Specific volume is the volume of one pound.

The specific volume of water is $\frac{1}{62.4} = 0.016$ cubic feet, 62.4 pounds being the weight of a cubic foot of water at 62°.

The volume of a pound of dry steam, represented by the letter s, varies with the pressure.

The *British thermal unit,* B. t. u., is the amount of heat necessary to raise one pound of water from 62° F. to 63° F.

The *heat of the liquid* is the amount of heat expressed in B. t. u. necessary to raise one pound of water from 32° to the temperature desired.

If the specific heat of water was unity throughout the entire range of temperature, the heat of the liquid would be 32 less than the temperature.

The specific heat is slightly above unity at some temperatures, and slightly below unity at other temperatures.

The heat of the liquid is represented by the letter q.

Relation of pressure and temperature of saturated steam:

Regnault found that the temperature of steam depended upon the pressure; that the temperature of the steam, if it was not superheated, was exactly the same as that of the water in contact with it.

Particles of water may float in steam the same as fog floats in the air. This does not affect the temperature of *the steam.*

Total latent heat or *heat of vaporization* (represented by r):

If a pound of water at 32° is heated up to the boiling point at atmospheric pressure, 180.3 heat units (the value q of the heat of the liquid), will be consumed in raising the temperature from 32° to 212°. If now heat is added, the water will gradually go off as steam. To entirely vaporize this pound 969.7 B. t. u. will be needed. This 969.7 B. t. u. is the heat of vaporization at this pressure. It is often called the total latent heat.

The heat of vaporization is different at different pressures.

The total heat at any pressure (represented by the Greek letter λ called lambda) is the amount of heat necessary to raise a pound of water from 32° to the temperature corresponding to the pressure and to then entirely vaporize the water at this temperature and under the constant pressure.

It is evidently equal to the sum of q and r, the heat of the liquid and the total latent heat.

Until recently the values of λ as determined by Regnault were used, although they were known to be somewhat in error; recently, however, Mr. H. N. Davis has deduced correct values of the total heat by making use of the experimental data at hand, mainly that of Grindley, of Griessman, and of Knoblauch, all three of whom were at work on the determination of the values of the specific heat of superheated steam.

Heat equivalent of external work and *heat equivalent of internal work:*

As water passes into steam at constant pressure and at constant temperature, we have seen that the heat of vaporization r is required.

This value r is made up of two parts. One part, the heat equivalent of the external work, can be calculated.

The other part, the heat equivalent of the internal work, is obtained by subtracting the first from r.

The following example will illustrate the method of calculating the heat equivalent of the external work.

Suppose one pound of water at 32° to be placed in the bottom of a vertical cylinder of one square foot piston area. Let the piston be weighted so that together with the atmospheric pressure there is a load of 100 pounds per square inch on the piston.

If heat is added to the water, vaporization will not begin till a temperature of 327.86° F. is reached. As vaporization takes place the piston rises in the cylinder.

When all the water has been made into steam, the piston will stand 4.432 feet above the bottom of the cylinder.

The pound of water occupied 0.016 of a cubic foot, and as the cylinder is one square foot in sectional area, the piston must have moved up a distance of $4.432 - 0.016 = 4.416$ feet.

The external work done is $100 \times 144 \times 4.416$ foot-pounds. Dividing this by 778, the mechanical equivalent of one heat unit, gives as a result 81.9 B. t. u.

The heat equivalent of the external work is represented by $A p u$. A is the heat equivalent of one foot-pound and is equal to $\frac{1}{778}$; p is the absolute pressure on the square foot; u equals the change in volume in passing from water to steam.

Subtracting the heat equivalent of the external work from the total latent heat gives the heat equivalent of the internal work. This is represented by the Greek letter ρ (called rho).

In this case, at this pressure, ρ amounts to

$$887.6 - 81.9 = 805.7 \text{ B. t. u.}$$

This heat equivalent of the internal work increases as the volume of a pound of steam increases.

As the volume occupied by a pound of steam at very low pressures is large, it will be found that the internal latent heat is a large proportion of the total heat.

Where this heat goes to, may be illustrated thus.

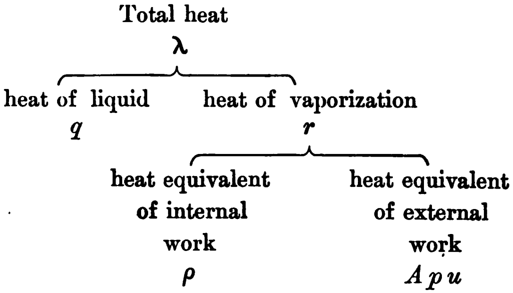
All substances are supposed to be made up of small particles called molecules.

The pound of water occupying 0.016 of a cubic foot had a certain number of these molecules. The number remained the same in the pound of steam which filled a volume of 4.432 cubic feet.

The relative distance between these molecules has been increased 275 times.

Each molecule exerts an attraction for its neighbor, and as this attraction has been overcome through space, work has been done. This work has required an equivalent expenditure in heat.

This internal work is often called *disgregation work.*


Total heat
λ
heat of liquid
q
heat of vaporization
r
heat equivalent of internal work
ρ
heat equivalent of external work
Apu

Volume of a pound of mixture of steam and water:

The volume of a pound of steam is s, the volume of a pound of water is 0.016 of a cubic foot.

If the mixture is x parts steam by weight the volume v of the pound of mixture is

$$v = xs + (1 - x)\,0.016$$
$$v = x(s - 0.016) + 0.016$$

Total heat of a pound of mixture of steam and water above 32°:

Before any water can be made into steam at a given pressure, the whole of the water must first be heated to the temperature corresponding to the pressure. Then if x parts by weight are made into steam, the heat $x\,r$ must be added, making the total heat to be added $q + x\,r$.

Total heat of a pound of a mixture of steam and water above any given temperature.

First find the heat of the pound of mixture above 32° equal to $q + x\,r$, then subtract the heat of the liquid at the given temperature.

Superheated steam is steam of a higher temperature than that corresponding to saturated steam of the same pressure.

The difference of temperature is the number of degrees of superheating.

To tell whether or not steam is superheated, a thermometer, a steam gage, and a table or plot giving the temperatures of saturated steam are needed.

Knoblauch, Linde, and Klebe, from recent experiments made in Munich, have determined the following equation for superheated steam:

$$pv = 85.85\ T - p\ (1 + 0.00000976p)\ \left\{ \frac{150{,}300{,}000}{T^3} - 0.08328 \right\}$$

A much more simple equation giving results agreeing with the above within 0.8 of one per cent is

$$pv = 85.85\ T - 0.256\ p$$

where p is the absolute pressure in pounds on the square foot, and v is the volume of one pound. T is the absolute temperature of the superheated steam in degrees F.; this is found by adding 459.5 to the temperature of the steam as given by the thermometer.

Having the temperature and pressure as known terms, the volume may be found, or, with a known volume and a known pressure, the temperature may be found.

Specific heat of superheated steam at constant pressure $= C_p$.

It was formerly assumed that this specific heat of superheated steam was constant. It is now known that the specific heat increases with increase in pressure, but at any constant pressure the value decreases as the amount of superheating increases.

The mean values of the specific heat for different pressures and for different degrees of superheating, as given by the experimental work of Thomas & Short, are quoted in the accompanying table.

Mean Value of Specific Heat of Superheated Steam (Thomas & Short)

Degrees of Superheat Fahr.	Pressure, pounds per square inch, absolute					
	6	15	30	50	100	200
20	0.536	0.547	0.558	0.571	0.593	0.621
50	0.522	0.532	0.542	0.555	0.575	0.600
100	0.503	0.512	0.524	0.537	0.557	0.581
150	0.486	0.496	0.508	0.522	0.544	0.567
200	0.471	0.480	0.494	0.509	0.533	0.556
250	0.456	0.466	0.481	0.496	0.522	0.546

Total heat of a pound of superheated steam: This is evidently equal to the total heat of a pound of saturated steam of the same pressure, plus the average value of the specific heat for the range of superheating times the number of degrees of superheat.

Using the letters which represent the different values

$$q + r + C_p \times \text{(degrees superheat)} ;$$

or

$$\lambda + C_p \times \text{(degrees superheat)}.$$

At the back of the book there are two charts, one giving the different values of t, q, r, λ, $A\ p\ u$, ρ, and s for pressures from 0 to 10 pounds absolute and the other giving values of the same terms from 10 pounds absolute to 250 pounds absolute.

These curves, which are drawn to represent the values given in Peabody's Steam Tables, the tables in general use by engineers, will serve to give values with a moderate degree of accuracy. For accurate work such values should be taken from some reliable steam table which gives these values for each degree difference of temperature or for each pound increase in pressure.

Tables which give values for intervals of 5 pounds, and where values for intermediate points must be obtained by interpolation, are fairly accurate at high pressures, but unreliable at low pressures on account of the error due to interpolation. Even tables reading to one pound are unreliable at low pressures for the same reason.

If either Peabody's Steam Tables or the Steam Tables by Marks and Davis are used, all low pressure values should be taken from the temperature table which gives values for each degree from 32°. The pressure corresponding to each temperature is given also. As there are seventy sets of values for pressures between 0 and one pound absolute, sufficient accuracy can be obtained.

The charts show that the total heat and the heat equivalent of the external work change but little; that the temperature, the specific volume of steam, and the heat equivalent of the internal work (internal latent heat), change rapidly at low pressures and slowly at high pressures.

In order to show the application of the discussion in the preceding pages a few examples will be solved.

Problem (1). How much heat will it take to make 3 pounds of water at 60° F. into wet steam at 150 pounds *bsolute* pressure? The steam is primed 2 per cent.

If the wet steam contains 2 per cent moisture there must be 98 per cent dry steam.

In solving problems in steam where use is made of the values λ, q, r, s, ρ, $A\ p\ u$, etc., it must be remembered that these values are for one pound. It is advisable to work all problems as if the actual weight were one pound and to finally multiply the result by the actual weight.

The heat which must be added to a pound of water at 32° in order to make this into wet steam at this pressure is $q + 0.98r$ where q and r are the values of the heat of the liquid, and the total latent heat at 150 pounds absolute, respectively. The water was originally at 60°. The heat of the liquid of water at 60° must be subtracted from this to give the amount to be added per pound.

$$(q_{150\text{ lbs. abs.}} + 0.98r_{150\text{ lbs. abs.}} - q_{60^\circ\text{ F.}}) \times 3$$

From the chart and the table of heat of the liquid these values are

$$[330. + (0.98 \times 863.0) - 28.1] \times 3 = 3443.$$

Problem (2). What volume will the 3 pounds of wet steam occupy?

From the chart it appears that the volume of one pound of dry steam at 150 pounds absolute pressure is 3.0 cubic feet.

The volume of one pound of mixture or of wet steam is

$$v = 0.98\ (3.0 - 0.016) + 0.016 = 2.940$$

The 3 pounds will occupy a

$$\text{volume} = 3\ v = 8.820 \text{ cubic feet.}$$

Problem (3). An engine is supplied with steam at 144 pounds absolute pressure. The steam contains one per cent of moisture.

The engine uses 2,800 pounds of steam per hour (all through the cylinders, there being no jackets).

The indicated horse-power is 200. The temperature of the exhaust at the condenser is 126° F.

The air pump discharges the condensed steam back to the boilers through a primary heater on the exhaust pipe.

The temperature of the feed-water entering the boiler is 100° F.

Each pound of coal burned under the boilers gives up 14,500 B. t. u., 9,900 of which are taken up by the boiler and utilized in making steam.

What is the number of pounds of coal per horse-power as indicated?

What is the thermal unit consumption of the engine per horse-power per minute?

$$\left(q_{144\text{ lbs. abs.}} + 0.99r_{144\text{ lbs. abs.}} - q_{100^\circ\text{ F.}}\right)\frac{2800}{200}$$

gives the number of thermal units supplied by the boiler per I. H. P.

Substituting the values from the chart or tables

$$[326.7 + (0.99 \times 865.6) - 68.0] \times 14 = 15619.$$

Dividing this by 9900 gives the coal per I. H. P. of the engine alone as

$$\frac{15619}{9900} = 1.58 \text{ pounds.}$$

In calculating the thermal unit consumption of the engine it is customary to assume that the condensed steam could be returned by the air-pump to the boiler at the same temperature as that of the exhaust steam.

The thermal unit consumption per I. H. P. per minute is

$$\left(q_{144\text{ lbs. abs.}} + 0.99r_{144\text{ lbs. abs.}} - q_{126^\circ\text{ F.}}\right)\frac{2800}{200 \times 60} =$$

$$(326.7 + 857.0 - 94.0)\frac{7}{30} = 254.26$$

Problem (4). Suppose that the steam supplied to the ngine was of 144 pounds absolute pressure and 400° F. in *mperature; that* the steam consumption per hour was

2,600 pounds, and that the I. H. P. and other conditions were the same, what would be the B. t. u. per I. H. P. per minute?

$$\left\{ \lambda_{144\text{ lbs.}} + 0.587\,(400.0 - 355.29) - q_{126^\circ\text{ F.}} \right\} \frac{2600}{200 \times 60} =$$

$$\left\{ 1192.3 + 26.24 - 94.0 \right\} \frac{13}{60} = 243.65$$

The specific heat of superheated steam is taken from the preceding table as 0.587.

Should the engine be provided with steam jackets the weight of jacket steam per H. P. per minute times the B. t. u. given up by the condensation of one pound is to be added to the B. t. u. per H. P. per minute through the cylinders.

Problem (5). What is the thermal efficiency of the engines in (3) and (4) as previously explained?

$$\frac{33000}{778} = 42.42 \text{ B. t. u.}$$

$$\frac{42.42}{254.26} = 0.167 \qquad \frac{42.42}{243.65} = 0.174$$

Carnot engine. It is found in the preceding problem that the thermal efficiency of the engines is low.

One might be led to think that the steam engine was not as economical as it might be made to be. This is not the case, however. Many of our best engines when compared in thermal unit consumption with that of the theoretically perfect engine, working between the same pressures and temperatures, give 70 per cent comparative efficiency.

The theoretically perfect engine, called the Carnot engine, is not necessarily one with 100 per cent thermal efficiency, but one in which there are no losses from friction, conduction, radiation, etc. It is one in which all the heat supplied is accounted for by the sum of the heat withdrawn, and the heat transformed into work.

Evidently an engine to have 100 per cent thermal efficiency must transform all the heat it receives into and have none to throw away or be withdrawn. It can be shown that the efficiency of such a theoretically perfect engine is given by dividing the difference of temperature worked through in the cycle, by the absolute temperature at which heat was supplied to the engine.

A Carnot engine working through the same temperature intervals as those given in Problem (3) would have a thermal efficiency:

$$\frac{355.29 - 126.}{355.29 + 459.5} = 0.281$$

Comparing the actual with that of the Carnot:

$$\frac{0.167}{0.281} = 0.59$$

The thermal unit consumption per H. P. per minute for this case is with the Carnot engine:

$$\frac{42.42}{0.281} = 151.$$

The B. t. u. consumed per H. P. per minute by the actual engine and by the theoretical bear the same ratio as that of the thermal efficiencies:

$$\frac{151.}{254.26} = 0.59$$

The only correct way to quote the performance of an engine is by its thermal efficiency or by its B. t. u. consumption per I. H. P. per minute.

The weight of steam per H. P. per hour does not mean anything unless one knows the heat in that steam as supplied to the engine and the temperature and pressure of the exhaust.

One engine may develop a H. P. on 9 pounds of steam, the steam being highly superheated. Another engine with perhaps a higher thermal efficiency than the first may use ? *pounds per H. P.*

If two engines work under exactly the same conditions as to boiler pressure, steam, and vacuum, then a comparison may be made of the steam consumptions per H. P. per hour.

In the Carnot engine it is supposed that the same charge of working substance, air, steam, or whatever it may be, is alternately heated and cooled in the cylinder. The actual engine has a new supply of working substance brought into the cylinder on each power stroke.

It would seem better to compare the actual engine with a perfect engine which was similarly supplied. By a perfect engine is meant one in which there is no friction, no radiation, and no absorption or conduction of heat by the cylinder walls; one in which the expansion drops the pressure down to that of the back pressure. Such an engine is called a *non-conducting engine* or an engine working on the *Rankine cycle.*

Non-conducting engine. The amount of heat which must be added to a pound of feed water at the boiler to make it into a pound of steam of the condition as supplied to the engine, assuming that the feed water enters the boiler at the temperature of the engine exhaust, is $q_1 + x_1 r_1 - q_2$; where q_1 is the heat of the liquid, r_1 the total latent heat or heat of vaporization at boiler pressure, and x_1 is the quality of the steam made by the boiler. If there is one per cent priming in the steam then $x_1 = 0.99$. If the quality of the steam after an adiabatic expansion from cut-off down to the back pressure is x_2 the heat to be abstracted during the exhaust is $x_2 r_2$ where r_2 is the latent heat of steam at the pressure corresponding to the exhaust.

The efficiency of any engine is the difference between the heat supplied and the heat exhausted divided by the heat supplied. In this case the efficiency becomes

$$\frac{q_1 + x_1 r_1 - q_2 - x_2 r_2}{q_1 + x_1 r_1 - q_2} = 1 - \frac{x_2 r_2}{q_1 + x_1 r_1 - q_2}$$

the values so figured are plotted, a line marked liquid line will be obtained. From the plot the entropy of the liquid at any temperature may be read directly:

at 697.5° absolute the entropy of the liquid is 0.35
at 800° " " " " " " " 0.49
at 550° " " " " " " " 0.11

To make steam at a given pressure from water at 32° the water is first heated up to the temperature corresponding to the pressure by the addition of the heat of the liquid q. The entropy increases by an amount which may be read from the liquid line. Next the heat of vaporization r is added and the water gradually passes into steam at the same temperature. The increase in entropy due to the addition of the heat of vaporization is $\frac{r}{T}$. If the value of $\frac{r}{T}$ be figured for each pressure and laid off to the right of the liquid line, the dry steam line is obtained. If instead of vaporizing the entire pound of water, only 80 per cent of it had been vaporized, the heat added at constant temperature would have been $0.80\ r$ and the increase in entropy due to vaporization $\frac{0.80\ r}{T}$ or 80 per cent of the value between the liquid line and the dry steam line. The horizontal distance between the liquid line and the dry steam line has been divided into 10 parts marked $x = 0.10$, $x = 0.20$, etc., and these parts each subdivided into 5 additional parts.

Illustration. The entropy of a pound of dry steam at 800° absolute temperature is read from the chart as 1.58. The entropy of a pound of dry steam at 550° absolute is 2.02. The entropy of a pound of mixture of steam and water which is 80 per cent steam by weight at 550° absolute is 1.56.

Between the liquid line and the dry steam line there are four curves which are used in finding the absolute tempera*re corresponding* to any absolute pressure.

The absolute temperature of steam

at 2 pounds absolute pressure appears to be 587.5°
at 4 pounds “ “ “ “ “ 613.°
at 10 pounds “ “ “ “ “ 654.°
at 220 pounds “ “ “ “ “ 850.°

The entropy of a pound of mixture of steam and water at 50 pounds absolute pressure, the mixture being 34 per cent steam by weight, is read on the chart as 0.855.

Beyond the dry steam line are lines marked 250 pounds, 200 pounds, 150 pounds, etc., leading upward from the dry steam line. These lines give the entropy of superheated steam.

Take for illustration 150 pounds absolute. This line starts from the dry steam line at 818°, the absolute temperature of saturated steam at this pressure; as heat is added to the dry steam and the pressure kept constant, the temperature increases and the entropy increases. The temperature increases more rapidly than the entropy. As an illustration, the entropy of a pound of steam at 150 pounds absolute pressure, superheated 100° F., is 1.632.

The temperature of saturated steam at 150 pounds is 818° absolute.

The entropy of the liquid at 818° absolute is 0.513.

The entropy of a pound of dry steam at 818° is 1.565.

The increase in entropy due to the 100° superheat is .067.

During a reversible adiabatic expansion the entropy remains constant. This plot is a great help in solving for the final condition of a mixture after an adiabatic expansion. In Problem (6) on the non-conducting engine, steam at 144 pounds absolute pressure with one per cent priming was expanded adiabatically to 126° F. or 585.5° absolute. It was found by a numerical calculation that $x_2 = 0.79$. This value may be found at once by the chart. Follow along at the temperature level corresponding to 144 pounds until $x =$

0.99 is reached. The entropy at this point is 1.56. Follow down on entropy 1.56 to temperature 585.5 and note the value of x; x is found to be 0.79.

Problem (7). Steam at 100 pounds absolute pressure, superheated 125°, expands adiabatically to 10 pounds pressure absolute. What per cent is steam at the end of the expansion? At what pressure is the steam just dry; that is with no moisture and with no superheat? Follow up on the 100 pound superheat line till a point is reached 125° above the temperature at which this line starts from the dry steam line. Read the entropy at this upper point as 1.68. Follow down at constant entropy till the dry steam line is reached at a temperature of 725° absolute. At this temperature level the pressure is found from the pressure curves to be 38 pounds absolute; continue on entropy line 1.68 down to the temperature corresponding to 10 pounds and note x as 0.93.

FLOW OF STEAM THROUGH AN ORIFICE

The velocity of steam at 100 to 150 pounds pressure, issuing from an orifice into the air, is from 1,300 to 1,500 feet per second. The weight of steam discharged through an orifice with rounded entrance, having 150 pounds absolute pressure on the entrance side, will be the same in amount for any back pressure from 90 pounds absolute down. At first sight this does not seem reasonable. The pressure drops at what is called the throat or the smallest section of the orifice or nozzle to 0.6 the absolute entrance pressure, provided the back pressure is not over 0.6 of the entrance pressure. Under these conditions, as the throat pressure and the velocity at the throat are the same, the quantity discharged will remain constant during changes in back pressure from 0.6 of the boiler pressure down to zero.

same is true also for gases.

Measurement of Dry Steam by the Flow through an Orifice

An empirical formula known as Napier's or as Rankine's gives very accurate results.

The orifice should have a rounded edge at entrance.

W = the weight of steam flowing per second.

P_1 = the absolute pressure in pounds per square inch on the entrance side.

P_2 = the absolute pressure in pounds per square inch on the exit side.

A = area of the orifice in square inches.

Where P_1 is equal to or greater than $\frac{5}{3} P_2$

$$W = A\frac{P_1}{70}$$

Where P_1 is less than $\frac{5}{3} P_2$

$$W = A\frac{P_2}{42}.\left\{\frac{3\ (P_1 - P_2)}{2\ P_2}\right\}^{\frac{1}{2}} = 0.0292\,A\ (P_1 P_2 - P_2^2)^{\frac{1}{2}}$$

As P_2 approaches P_1 more steam goes through the orifice than this formula gives.

This second formula is not to be recommended as accurate within 8 per cent when $\frac{P_2}{P_1}$ bears the ratio 0.85 or higher.

Design of a Turbine Nozzle for Complete Expansion

By gradually increasing the diameter of a nozzle beyond the throat or smallest section, the velocity of the steam in the nozzle may be increased as the pressure drops, till at the end of the nozzle a velocity of from 3,600 to 4,000 feet per second may be realized if the back pressure is low.

By complete expansion is meant a drop in pressure in the nozzle from the highest to the lowest pressure; that is there is no drop after leaving the nozzle.

The *method* commonly used in calculating a nozz given in the following pages, but the derivation of the formulæ used is omitted.

Let the subscript i denote conditions and values of the high pressure steam at entrance to the nozzle; the subscript t similar conditions at the throat, and the subscript e at exit.

$H_i = q_i + x_i r_i$ for saturated steam.

$H_i = q_i + r_i + C_p$ (degrees of superheat) for superheated steam.

$H_t = q_t + x_t r_t$ for saturated steam.

$H_t = q_t + r_t + C_p$ (degrees superheat) for superheated steam.

$H_e = q_e + x_e r_e$.

The values of x_t and x_e are read from the temperature entropy plot, assuming adiabatic expansion from the condition x_i. If the steam is superheated to start with, the chart is used in the same way after locating the starting position. Call V_t the velocity at the throat in feet per second and V_e the velocity at the exit.

$$V_t = 224 \sqrt{(H_i - H_t)}$$

$$V_e = 224 \sqrt{0.85(H_i - H_e)}$$

The area of the throat and the exit sections are calculated thus: The volume of one pound of steam at the throat is

$$v_t = x_t (s_t - 0.016) + 0.016$$

where s_t is the volume of one pound of dry steam at the throat pressure. Should the steam be superheated at the throat, the volume of a pound would be calculated by the formula given in the earlier part of this chapter.

$$\frac{v_t \times \text{weight per second}}{V_t} = \text{area of throat in square feet.}$$

In finding V_e, 85 per cent of $H_i - H_e$ was used because a friction loss amounting to 15 per cent of $H_i - H_e$ was assumed to occur in the nozzle. The friction loss up to the throat is small and is not considered in this calculation. A small allowance is sometimes made for it, however.

The effect of this friction and of the conduction of heat by the nozzle is to make the steam more nearly dry at exit than it would have been after an expansion at constant entropy. The increased dryness may be found by $\frac{0.15\,(H_i - H_e)}{r_e}$.

This added to x_e gives x_f the final condition leaving the nozzle.

$$v_e = x_f\,(s_e - 0.016) + 0.016$$

$$\frac{v_e \times \text{weight per second}}{V_e} = \text{area of exit in square feet.}$$

Problem (8). A de Laval turbine of 350 H. P. rated capacity is supplied with seven nozzles. The pressure of steam at entrance is 200 pounds absolute, the steam being superheated 35°. The exit pressure is 2 pounds absolute. Assume the friction loss in the nozzle to be 15 per cent. Assume also that 65 per cent of the kinetic energy of the steam is utilized by the wheel. Find steam per H. P. per hour and the diameters of each nozzle at exit and at the throat.

Refer to temperature entropy chart, 200 pounds pressure, 35° superheat. The entropy is 1.57. Follow down on 1.57 until the temperature corresponding to 0.6 × 200 = 120 pounds pressure is reached. Read $x_t = 0.99$. Continue on 1.57 until the temperature corresponding to 2 pounds is reached. Read $x_e = 0.80$.

$$H_i = 354.3 + 843.5 + (0.60 \times 35) = 1218.8$$

$$H_t = 312.3 + (0.99 \times 876.9) = 1180.4$$

$$H_e = 94.2 + (0.80 \times 1021.9) = 911.7$$

$$V_t = 224\sqrt{1218.8 - 1180.4} = 1393$$

$$V_e = 224\sqrt{0.85\,(1218.8 - 911.7)} = 3618$$

$$\frac{0.15\,(1218.8 - 911.7)}{1021.9} = 0.045$$

$$x_f = x_e + 0.045 = 0.80 + 0.045 = 0.845$$

The kinetic energy per pound of a jet issuing with a velocity of 3,620 feet per second is $\frac{3620 \times 3620}{2 \times 32.2}$. As 65 per cent of this is utilized, the energy received by the wheel per second per pound of steam is

$$\frac{0.65 \times 3620 \times 3620}{64.4} \text{ foot-pounds.}$$

The energy needed per second to develop 350 H. P. is

$$350 \times \frac{33000}{60}$$

hence the number of pounds of steam which must be supplied per second is

$$\frac{350 \times 33000 \times 64.4}{0.65 \times 3620 \times 3620 \times 60} = 1.455$$

The steam per H. P. hour is

$$\frac{1.455 \times 3600}{350} = 14.96 \text{ lbs.}$$

The steam per nozzle per second is

$$\frac{1.455}{7} = 0.208 \text{ lbs.}$$

The volume of a pound of mixture at the pressure and the condition at the throat is $0.99 (3.723 - 0.016) + 0.016 = 3.686$ cubic feet. The volume of a pound at exit is $0.845 \times (173.1 - 0.016) + 0.016 = 146.2$ cubic feet. The area of the throat in square feet is

$$\frac{3.686 \times 0.208}{1390}; \text{ or } 0.32 \text{ inches diameter.}$$

The area of the nozzle at exit in square feet is

$$\frac{146.2 \times 0.208}{3620}; \text{ or } 1.33 \text{ inches diameter.}$$

CALCULATING THE SIZE OF A STEAM MAIN

The indicator when applied to the steam chest as explained in Part II, page 107, sometimes shows that the *steam pipe is not large* enough to supply the engine.

If the pipe is furnishing steam to a slow speed engine, and the pipe is not much under the correct size, a drum placed in the steam pipe close to the engine may remedy the trouble. The volume of this drum should be at least four times the volume of the cylinder.

If the pipe supplying a high-speed engine is too small, it will have to be changed in order to remedy the trouble.

In figuring the size of a steam pipe or steam main it is customary to allow 6,000 feet velocity of the steam per minute if the pipe is short, with but few elbows. If the pipe is of moderate length, 5,000 feet per minute; 4,000 is used on long runs where there are many elbows and bends.

Knowing the weight of steam to be carried through the pipe per minute, and knowing also the *lowest pressure* at which the plant will ever work, the volume of the steam can be figured. This volume divided by the allowable velocity will give the area of the pipe needed.

Example: 300 pounds of steam per minute are to be carried through the pipe. The highest pressure at which the plant runs is 125 pounds absolute. The lowest, 100 pounds absolute.

From the chart it is found that the volume of one pound of steam at 100 pounds pressure is 4.4 cubic feet.

$$\frac{4.4 \times 300}{5000} = 0.264 \text{ sq. ft.} = 38.01 \text{ sq. in.}$$

This area corresponds to 6.96 inches diameter.

For the higher pressure an area of

$$\frac{3.6 \times 300}{5000} = 0.216 \text{ sq. ft. is needed.}$$

If the pipe has this area the velocity of steam through the pipe at the lower pressure will be

$$\frac{4.4}{3.6} \times 5000 = 6100 \text{ feet per minute.}$$

PERFECT GASES

The characteristic equation of a perfect gas or the equation giving the relation between the absolute pressure, the volume and the absolute temperature is

$$\frac{Pv}{T} = \frac{P_1 v_1}{T_1} = \frac{P_2 v_2}{T_2}, \text{ etc.}$$

This relation was determined experimentally.

The volume of a pound of air at atmospheric pressure and at freezing point has been determined experimentally to be 12.39 cubic feet. That of hydrogen, 178.2 cubic feet.

Atmospheric pressure is 14.7 pounds on the square inch, or 2116.3 pounds on the square foot, equivalent to 29.92 inches of mercury or 760 mm. of mercury. The temperature T is absolute; as has been stated, this is found by adding 459.5 to the reading of a Fahrenheit thermometer.

A few examples will best illustrate how use is to be made of this equation.

(1). What will be the volume of one pound of air at 100 pounds absolute pressure and at 139.3° F.?

$$\frac{14.7 \times 12.39}{491.5} = \frac{100 \times v}{459.5 + 139.3}$$

$$v = 2.22 \text{ cu. ft.}$$

(2). What will be the weight of a cubic foot of air at this pressure and temperature?

$$\frac{1}{v} = \frac{1}{2.22} = 0.45 \text{lbs.}$$

(3). An air compressor draws in 100 cubic feet of free air per minute at 14.6 pounds pressure (absolute) and at 60° F. The air is compressed to 200 pounds absolute and leaves at 120° F. What is the volume of the air discharged?

$$\frac{14.6 \times 100}{459.5 + 60} = \frac{200 \times v}{459.5 + 120}$$

$$v = 8.14 \text{ cu. ft.}$$

(4). A balloon of 10,000 cubic feet capacity, weighing together with car, sand bags, etc., 550 pounds, has 9,000 cubic feet of hydrogen run into it at 80° F. and at 30.2 inches of mercury pressure, this being the temperature and the pressure of the surrounding air. Find the weight of gas run in; the pull on the rope holding the balloon to the ground, and the amount the balloon would have to be lightened in order for it to reach a height where the barometer reads 20 inches and the temperature is 32° F. What would be the pressure of the gas on the inside of the balloon at the upper level, assuming that no gas escapes?

The lifting force is the difference between the weight of the air displaced by the hydrogen and the weight of hydrogen run in.

Call V_H the volume of one pound of hydrogen at the pressure and temperature at the ground, and V_A that of a pound of air at the same place.

$$\frac{29.92 \times 12.39}{491.5} = \frac{30.2 \times V_A}{539.5}; \qquad V_A = 13.45 \text{ cu. ft.}$$

$$\frac{29.92 \times 178.2}{491.5} = \frac{30.2 \times V_H}{539.5}; \qquad V_H = 193.2 \text{ cu. ft.}$$

$$\frac{9000}{13.45} - \frac{9000}{193.2} = 669 - 46.6 = 622.4 \text{ lbs.}$$

$622.4 - 550 = 72.4$ pounds pull needed to hold the balloon to the ground.

As the balloon rises the hydrogen expands and fills the balloon; and as the balloon continues to rise, the hydrogen if not allowed to escape would produce a pressure tending to rupture the balloon.

The weight of air displaced by the balloon at the upper level is calculated thus:

$$\frac{29.92 \times 12.39}{491.5} = \frac{20 \times V^1_A}{491.5}; \qquad V^1_A \quad 18.52 \text{ cu. ft.}$$

The 10,000 cubic feet of air displaced weigh

$$\frac{10000}{18.52} = 539.5 \text{ lbs.}$$

As no hydrogen has escaped, according to the assumption, its weight is 46.6 pounds, as found previously.

$539.5 - 46.6 = 492.9$

$550 - 492.9 = 57.1$ lbs., the weight of sand which must be thrown out in order to reach this level.

The pressure P inside of the balloon at the upper level is

$$\frac{30.2 \times 9000}{539.5} = \frac{P \times 10000}{491.5}; \qquad P = 24.7 \text{ inches.}$$

The outside air pressure is 20 inches, so the excess pressure inside the balloon is 4.7 inches of mercury or 2.35 pounds per square inch approximately.

Measurement of air by the flow through an orifice:

Experiments have shown that the following empirical formula gives quite accurate results for orifices up to one inch in diameter.

The orifice should be made with a rounded entrance, the radius of the curve being equal to the diameter of the orifice, and the length of the straight part of the orifice should be equal to the diameter.

Where the pressure on the entrance side of the orifice is greater than twice the pressure on the exit side:

$$w = 0.530 \frac{P}{\sqrt{T}} a$$

where w is the weight of air per second. P is the absolute pressure on the square inch on the entrance side. T is the absolute temperature of the air on the entrance side. a is the area of the orifice in square inches.

Isothermal line. The equation for an isothermal expansion or compression of a perfect gas is $Pv = P_1 v_1$, or the product of the absolute pressure and the volume is a constant. The work required for an isothermal compression, or developed by an isothermal expansion of a gas is

$$W = 144 P_1 v_1 \times 2.3026 \log. \frac{P_2}{P_1}$$

where W is in foot-pounds. P_1 is the absolute pressure on the square inch at the beginning of compression or at the end of expansion; v_1 is the volume in cubic feet at this pressure. P_2 is the absolute pressure on the square inch at the end of an isothermal compression or at the beginning of an isothermal expansion. The heat which must be abstracted during an isothermal compression or added during an isothermal expansion is $\frac{W}{778}$.

Adiabatic line. The equation in terms of pressure and volume representing an adiabatic expansion or compression of a perfect gas is

$$P\,v^{1.405} = P_1\,v_1^{1.405}$$

The temperature along an adiabatic change may be calculated by combining this equation with the characteristic equation for gases

$$\frac{P\,v}{T} = \frac{P_1\,v_1}{T_1}$$

The work done by an adiabatic expansion of a perfect gas or required for an adiabatic compression is

$$W = \frac{144\,P_1\,v_1}{0.405} \times \left[1 - \left(\frac{v_1}{v_2}\right)^{.405}\right]$$

where W is in foot-pounds. P_1 is the absolute pressure on the square inch at the beginning of expansion or at the end of compression; v_1 is the volume at this pressure. v_2 is the volume after the expansion or at the beginning of compression. The volumes are measured in cubic feet.

This value $\frac{v_1}{v_2}$ will always come out less than unity. Suppose for illustration $\frac{v_1}{v_2}=0.3$. This is to be raised to the 0.405 power. The logarithm of 0.3 is 9.47712 — 10. Write this 999.47712 — 1000 and multiply by 0.405 thus:

$$(999.47712 - 1000) \times 0.405 = (404.78823 - 405.) \text{ or}$$
$$9.78823 - 10$$

The number corresponding to this log. is 0.6141.

COMPRESSING AIR

The minimum work required to compress air and to deliver it at the temperature of the intake would be that needed for an isothermal compression

$$P v^{1} = P_1 v_1^{1}$$

For such a compression heat must be abstracted, as has been shown.

If no heat was abstracted during compression the compression becomes adiabatic and

$$P v^{1.405} = P_1 v_1^{1.405}$$

The ordinary compressor is water jacketed and although attempts are made to get an isothermal compression, the actual compression is neither isothermal nor adiabatic, but somewhere between these two lines. For the best compressors a line having the equation

$$P v^{1.2} = P_1 v_1^{1.2}$$

represents the compression. The compression in the average compressor is more nearly represented by a line having the equation

$$P v^{1.3} = P_1 v_1^{1.3}$$

The H. P. required at the compression cylinders of a two-stage compressor to compress a certain number of cubic feet of free air (that is, cubic feet of air as taken from the room) per minute, from the pressure at the end of the suction stroke to the delivery pressure along a line having the equation

$$P v^{1.3} = \text{a constant, is}$$

$$\frac{144 \times 2 \times P_s v \times 1.3}{33000 \times 0.90 \times 0.3} \left\{ \left(\frac{P_d}{P_s}\right)^{\frac{0.3}{2.6}} - 1 \right\}$$

which reduces to $0.0422\, P_s v \left\{ \left(\frac{P_d}{P_s}\right)^{.1154} - 1 \right\} = \text{H. P.}$

where P_s is the absolute pressure per square inch at end of the suction stroke, and P_d is the absolute delivery pressure; and v is the cubic feet of free air.

For a three-stage compressor this formula becomes

$$0.0633\, P_s\, v \left\{ \left(\frac{P_d}{P_s}\right)^{.077} - 1 \right\} = \text{H. P.}$$

The air at entrance to a compressor is slightly rarefied, thus making P_s less than atmospheric pressure, and the volume displaced by the piston of the compressor has to be greater than the free air on this account and also on account of the clearance. The ratio of the volume of free air per minute to the piston displacement is called the displacement efficiency. For the very best compressors with mechanically operated valves this is approximately 95 per cent. In the expression just given for calculating H. P., a value of 90 has been used for the displacement efficiency as this is more nearly correct for the ordinary compressor.

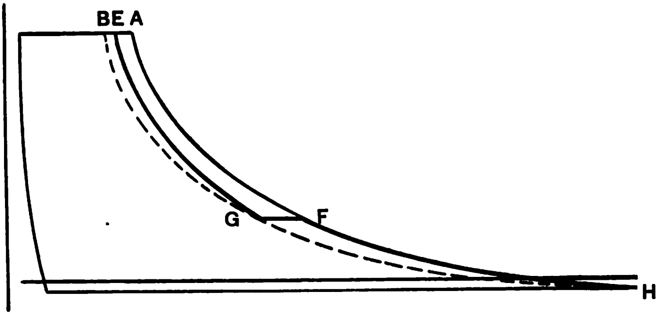


FIG. 58

The water jackets of a compressor are not very efficient in abstracting the heat of compression. By dividing the compression up between two or more cylinders or by compressing in stages it is possible to cool the air between stages down to its original temperature by passing it through inter-coolers placed between the compressor cylinders. This greatly reduces the work of compression as is shown by the illustration, Fig. 58.

The vertical line represents the cylinder head. H A represents the entire compression as taking place in one cylinder along a line $P v^{1.3}$ = a constant. The dotted line H B represents an isothermal compression. Air compressed along the line H A would shrink in volume from A to B as it cooled to its original temperature at intake. If the compression had been divided into two stages with an inter-cooler between the two cylinders the air delivered by the first stage at F would shrink in the inter-cooler to the volume at G before entering the second stage. The compression in the second stage is along a line G E of equation $P v^{1.3}$ = a constant, and the air at delivery shrinks from E to B. Evidently there has been a saving of work equal to that represented by the area F G E A. Against this saving is to be charged the extra mechanical loss due to the friction of the second cylinder. For a two-stage compression the absolute pressure on the square inch P_F is equal to $\sqrt{P_H \times P_A}$ the pressures at H and A being absolute. For a three-stage compression the absolute pressure at the end of the first stage should be $\sqrt[3]{P_H^2 \times P_A}$ and at the end of the second stage the pressure should be $\sqrt[3]{P_H \times P_A^2}$; the pressures P_H and P_A being absolute.

Problem on the compressor. 1,000 cubic feet of free air per minute are compressed to 265.3 pounds gage pressure in a two-stage compressor which is steam driven. The displacement efficiency of the compressor is 90 per cent. The pressure in the first cylinder at the end of the suction stroke is 14.0 pounds absolute, or 0.7 pounds below the atmosphere. The compression is along a line $P v^{1.3}$ = a constant. Calculate the H. P. needed at the compressor cylinders, and *assuming a mechanical* efficiency of 85 per cent, what is the

I. H. P. needed at the steam cylinders? What should be the pressure at the end of the compression in the first stage?

$$\text{H. P.} = 0.0422 \times 14. \times 1000 \left\{ \left(\frac{265.3 + 14.7}{14.0} \right)^{.1154} - 1 \right\} = 244.$$

$\frac{244}{0.85} = 287$ H. P. for steam cylinders. Absolute pressure at end of first stage $= \sqrt{280 \times 14} = 62.6$; or 47.9 pounds gage pressure.

PROBABLE HORSE-POWER OF AN ENGINE

The method of finding the number of expansions was explained in Chapter I, at page 11. As far as the power developed by a compound or triple expansion engine is concerned, the total number of expansions worked through by the steam might be made in the low pressure cylinder. By dividing the expansion between two or three cylinders the total cylinder condensation is reduced and a better rotative effect is obtained.

The expansion line of an indicator card does not vary much from a rectangular hyperbola having the equation $Pv =$ a constant, and such an equation is commonly assumed.

The entire cycle of expansions being assumed to take place in the low pressure cylinder, the M. E. P. for the entire cycle is figured for the low, and this M. E. P. multiplied by a constant between 0.7 and 0.9, which makes allowance for the losses in area and in M. E. P. due to the rounding of the card at cut-off and at release, and to the loss due to compression. For a Corliss valve gear these losses are small and the multiplier 0.9 would be used. For a plain slide valve a multiplier as low as 0.7 would be used.

The expression for M. E. P. is

$$\text{M. E. P.} = \frac{P_1}{n} + \frac{P_1}{n} \times 2.3026 \text{ log. } n - P_2$$

cylinder up to cut-off and up to release. This is called the total weight of mixture.

The weight of steam in the cylinder at compression is found by assuming that the space between the piston and the head of the cylinder, including port passages, is filled with dry steam of the absolute pressure at compression.

The weight at compression is equal to the (per cent clearance plus the per cent compression) times the piston displacement and times the weight of a cubic foot of steam at the absolute pressure at compression. (The weight of a cubic foot of steam is the reciprocal of the volume of a pound.)

Call this weight at compression M_0.

The weight of mixture in the cylinder per stroke is $M + M_0$.

The volume at cut-off is equal to the (per cent clearance plus the per cent of cut-off) times the piston displacement. Call this V_1.

This volume is filled by $(M + M_0)$ pounds of mixture at the absolute pressure of steam at cut-off as obtained from the diagram.

It has previously been shown (page 127) that the volume of one pound of mixture is $V = x\ (s - 0.016) + 0.016$ where x is the per cent steam by weight.

Hence V_1, the volume of $(M + M_0)$ pounds, must equal $V_1 = (M + M_0)\ x\ (s - 0.016) + (M + M_0)\ 0.016$. The volume s of one pound of steam can be found from tables or from a chart.

x, the only unknown term, is obtained by solving this equation.

The percentage of the mixture accounted for as steam at release is found in a similar way. The volume V_1 is replaced by V_2, the volume including clearance and the piston displacement up to release.

s *is taken at the* absolute pressure at release.

The percentage of mixture accounted for as steam at release frequently is as low as 0.6.

The indicator by itself does not show that there is any water with the steam in the cylinder. If the steam consumption of an engine is figured from the indicator card, assuming dry steam at release and dry steam at compression, a result as low as 18 pounds may be obtained when the actual consumption, as measured by the steam condensed in the condenser, is as much as 30 pounds.

CHAPTER III

DESIGN OF A PLAIN SLIDE VALVE

Valve Setting on a Plain Slide Valve Engine and on a Corliss Engine

Cards taken from a steam engine often show defects, which may be due to an improper setting of the valve, as shown in Part II, or to a poor design of the valve gear.

A thorough knowledge of the plain slide valve may be best obtained by studying some one of the graphical methods used in designing such valves. The two graphical diagrams most generally used are the Zeuner and the Bilgram.

In nearly all stationary engines the cross-head is connected to the crank by a connecting rod. This connecting rod is about seven times the length of the crank. In some marine engines the connecting rod is only three and one-half times the length of the crank.

It is evident that when the crank turns from 0° to 90° on the forward stroke, the piston moves from the head end center towards the crank end a distance greater than one-half its entire travel. On the return stroke, starting at the crank center, the piston moves through less than half its travel during the first 90° angular motion of the crank.

If instead of a connecting rod, a slotted cross-head had been used to connect the cross-head to the crank pin, then the same angular motion of the crank, from either center, would make the same displacement of the cross-head from either end.

If the crank rotates with a uniform angular velocity the slotted cross-head is said to move in *harmonic motion.*

The smaller the ratio of the length of the connecting rod to that of the crank the greater is the variation from harmonic in the displacement of the cross-head.

An eccentric and an eccentric rod are equivalent to a crank and a connecting rod; the length of the crank being the distance from the center of the shaft to the center of the eccentric, called the eccentricity.

An eccentric is simply a crank with a crank pin so large that it takes in the shaft. The diameter of the eccentric has nothing to do with the travel given by it; this depends solely on the eccentricity.

The eccentric is generally set from 95° to 110° ahead of the crank. The excess over 90° is called the *angular advance.*

The eccentric rod is so long in proportion to the eccentricity of the eccentric that the travel of the valve is not affected appreciably by the slight angular movement of the eccentric rod and it is customary to assume that the valve moves in harmonic motion.

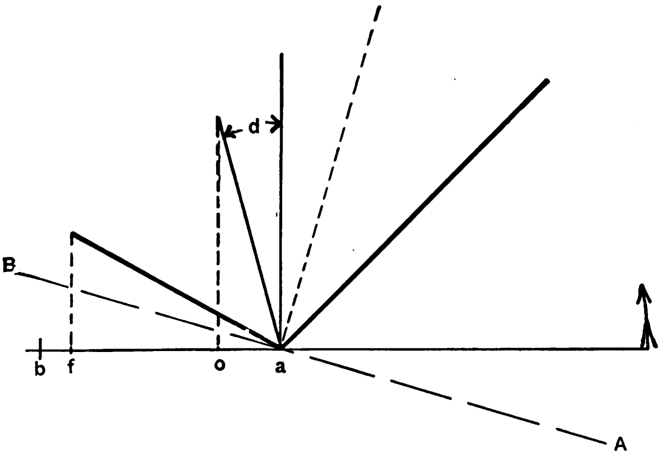


FIG. 59

In Fig. 59 the light lines show the crank at the center and the eccentric set with an angular advance d.

If the displacement of the valve is measured from the middle of its travel it is evident that the valve is displaced to the left an amount *a o*.

The maximum displacement to the left will be *a b*, equal to the eccentricity, and will occur when the crank is at the dotted line or an angle *d* before the 90° point.

The maximum displacement of the valve to the right comes at a crank position directly opposite to this dotted line. The valve has no displacements or is in mid position when the crank is at the position shown by the dash lines *A* at position making an angle *d* before each dead point.

Starting at the crank position *a A* the valve is in mid position and has no displacement; as the crank moves in the direction of the arrow the valve is displaced to the left, reaching a maximum *a b* when the crank is at the dotted line; beyond this crank position the displacements of the valve grow less till at the crank position *a B* the valve is back in mid position. While the crank is turning from *B* to *A* the valve is displaced in a similar manner on the right-hand side of mid position.

The displacement of the valve for the position shown by the heavy line is *a f*.

The following graphical solution, known as the *Zeuner Diagram*, will give the displacement of the valve for any crank angle.

The actual position of the crank and eccentric when the engine is on one dead center is shown by the full lines. In this graphical solution the angle *d* is laid off back of the 90° line, and on this line two circles are drawn, each circle being made of a diameter equal to the eccentricity of the eccentric.

The displacement of the valve from mid position for any crank angle is equal to the chord cut from either of these circles which the crank position may intersect. For example, at the crank position O S, shown by the dotted line, the displacement of the valve is O S to the left of the mid position. The maximum displacement comes at a crank angle when the chord cut is of maximum length or at angle *d* before each of the 90° points.

The valve has no displacement or is in mid position when there is no chord cut, or at the crank position O M making an angle *d* before each of the dead points.

These positions agree with those as found in the previous discussion.

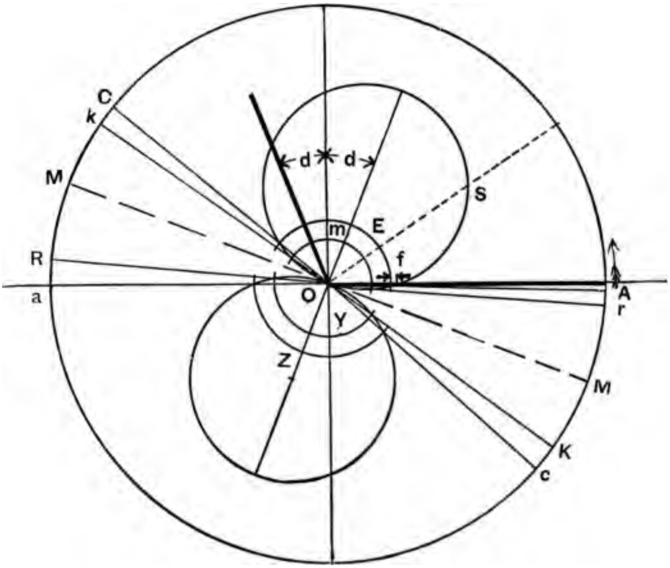


FIG. 60

Starting at a crank position O M (360° – *d*) the valve moves from mid position to the left, reaching a maximum distance to the left at a crank angle (90° – *d*); the displacements gradually grow less as the valve comes back to mid position, which is reached again when the crank is at O M

(180° — d). From this point on, the valve is similarly displaced to the right of its mid position.

A section through a plain slide valve and its seat is given below. It is seen that when the valve is in mid position, or in the center of its travel, the outer edge of each end of the valve overlaps the outer edges of the ports. This overlap is called the outside lap. It may or may not be the same on the two ends of the valve. The outer edges of the valve govern the admission of steam and the cut-off of steam.

The inner edges of the valve overlap the inner edges of the port in the same way. This distance is called the inside lap. The inside edge of the valve governs the release of steam and compression of steam.

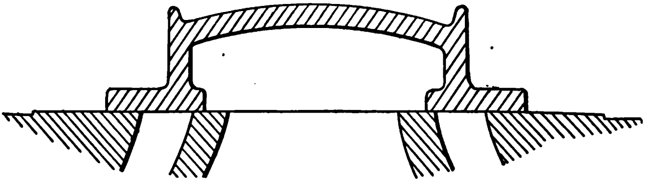


FIG. 61

Considering now the Zeuner diagram and the valve. When the crank is at the position O M (360° — d) the valve is in mid position, as shown in Fig. 61. As the crank reaches the position O A the valve is displaced to the left a distance O E. The arc O E is drawn with a radius equal to the outside lap on the right-hand end of the valve. At this crank position the outer edge of the valve is on the outer edge of the port and admission of steam is about to begin; see Part I, Chapter I, page 6. When the crank gets to the dead point, the displacement of the valve is greater than the outside lap by an amount f, which is called the *lead*. This lead varies from 0.01 of an inch to 0.38 of an inch in *different engines.*

When the crank gets to an angle (90° — *d*) the valve is displaced its maximum amount to the left; the outer edge of the valve is now to the left of the outer edge of the port by a distance equal to the eccentricity minus the outside lap.

The valve then begins to move back to its mid position and at a crank angle O C the outer edge of the valve is on the outer edge of the port, since the displacement is equal to O E, the outside lap, and cut-off occurs. It is seen that the displacement of the valve is the same at cut-off and at admission; at admission the displacements are increasing and at cut-off decreasing.

After cut-off, the steam in the cylinder expands as the valve moves to mid position at the crank angle O M = (180° — *d*) and then to the right of the mid position till the crank angle O R is reached.

At the crank angle O R the valve is displaced to the right of the mid position a distance O Y, which is equal to the inside lap on the right-hand end of the valve. (*Note.* This is drawn out of proportion on the diagram in order to avoid confusion.) The inner edge of the valve is now on the inner edge of the port and steam is about to escape from the cylinder over the bridge into the exhaust port. This is release. Steam is exhausted from the cylinder from the crank position O R to the crank position O K when the inner edge of the valve is again on the inner edge of the port, giving compression.

The displacement of the valve at release and at compression is the same in amount; the displacements at release are increasing and those at compression decreasing.

Consider now the left-hand end of the valve, which controls the distribution of steam to the left-hand end of the cylinder.

Evidently the valve must be displaced to the right for admission and cut-off and this displacement must be equal in amount to the outside lap. The arc o z is drawn with a radius equal to this outside lap. Admission comes at the dead point. There is then no lead. Cut-off comes at *o c*.

Release and compression occur when the valve is displaced to the left of the mid position an amount *o m* (drawn out of proportion) equal to the inside lap. This brings release at *o r* and compression at *o k*.

If the angular advance of the eccentric is decreased, or the angle *d* made less, admission, cut-off, release, and compression all come later in the stroke, as was shown by Figs. 27 and 28, Part II, page 103.

If the angular advance is increased, or the angle *d* made larger, all the events come sooner, each event moving ahead through the same crank angle.

The effect on the laps, both inside and outside, of a change in the length of the valve spindle, was pointed out in Part II, at page 104. The diagram makes it possible to trace out the effect of all such changes.

Suppose it is desired to increase the power of the engine. The cut-off may be lengthened by decreasing the outside lap and the release delayed by increasing the inside lap. Decreasing the outside lap would make admission come sooner and increasing the inside lap would make compression come earlier. To prevent these from coming abnormally early, the angular advance of the eccentric should be decreased.

If the outside lap and the inside lap are made zero, so that the length of each foot of the valve is just equal to the width of the steam port, compression and admission will come at the crank position O M and cut-off and release will come at the crank position O M opposite.

If the inside laps O M and O Y are reduced, the release comes sooner and the compression later. If these are made zero, release and compression come 180° apart at O M and O M. If it is desired to make the compression come still later, the inner edge of the valve may be cut back from the ver *edge of* the port, making a clearance. Evidently, when a valve has a clearance on its inner edge, to bring this edge to the inner edge of the port the valve must be displaced to the same side of the mid position that it was for cut-off. This means that the arc drawn on the Zeuner diagram to represent the inside clearance would be on the same side as that for the outside lap.

To lay out the seat for a valve: The width of the steam ports can be figured by assuming a velocity of steam as 100 feet per second, and the length of the port as about 0.8 the diameter of the cylinder.

Begin at the end of the valve which has the smaller outside lap. Starting at the outer edge of the port, measure off the outside lap. This outer edge of the valve will move to the right and to the left a distance in each direction equal to the eccentricity.

To allow the valve to over-travel its seat, the seat is depressed from a point about $\frac{1}{8}$ inch inside of the extreme travel out to the ends of the chest.

The travel of the valve in the other direction brings the outer edge over onto the bridge. The inner edge of the bridge should be about $\frac{1}{8}$ inch beyond this point. The inner edge of the bridge being now determined, and the width of the steam port having been figured, the width of the bridge itself is known.

To find the width of the exhaust port: Lay off the greater inside lap from the inner edge of the port onto the bridge. It may happen that the greater inside lap does not come on this end of the valve. The correct lap may be put on later after the width of the exhaust port is found. Measure from this lap towards the center of the chest a distance equal to the eccentricity, then add the width of the steam port; from this point, which is the inside edge of the bridge of the other end, lay off the bridge and then the steam port. Now lay out the laps.

SETTING A PLAIN SLIDE VALVE

A plain slide valve is almost always set for equal lead. The amount of lead and the direction of motion of the engine depend upon the position of the eccentric. The equality of the lead depends solely upon the length of the valve spindle.

Put the engine on a dead center. Remove steam chest cover. Loosen the set screws holding eccentric to shaft. Turn the eccentric till the steam port on one end is open its maximum amount. Caliper this distance from the outer edge of the port to outer edge of the valve. Now turn the eccentric until the maximum port opening on the other end is reached. Caliper this distance. Lengthen or shorten the valve spindle one-half the difference between these two measurements as taken by the calipers, lengthening if the head end opening was the greater and shortening if the crank end was the greater.

Turn the eccentric in the direction in which the engine is to run. As the engine is on the center the valve should move so as to open the port on that end of the cylinder and when the port has opened an amount corresponding to the lead desired, the eccentric should be set.

By setting a valve by this method it makes no difference whether or not there are bell crank levers or rockers between the eccentric and the valve.

SETTING CORLISS VALVES

The valves of a Corliss engine may be set quickest by the use of the indicator. If there is reason to suspect that the valve gear is very badly deranged, it might be well to see that the eccentric has a small angular advance and that the wrist plate swings through equal angles either side of the vertical. Should these angles be unequal, the length of the eccentric rod may be changed till equality is secured.

The engine is now started up and a set of cards taken. The release and compression are adjusted by lengthening or shortening the links between the wrist plate and the exhaust valves. Lengthening the link increases the exhaust lap and delays release and hastens compression; shortening it hastens release and delays compression. The cut-off is adjusted last. This is done by varying the length of the rods from the governor to the knock-off tappets and may be done while the engine is running. After setting the cut-off the engine should be tested with the governor pushed up against the top collar to see if the tappets keep the claws from engaging, and tested also with the governor at its lowest position, not resting on the starting block, to see if the safety tappets prevent the claws from catching.

EXPLANATION OF LOGARITHMS AND HOW TO USE THEM

The common logarithm of a number represents the power to which 10 must be raised to equal the number. Thus the log. 100 = 2 because 10 must be raised to the second power to equal 100. The log. 1000 = 3. The log. 10 = 1. The log. 1 = 0. Evidently the logarithm of a number between 1 and 10 will be between 0 and 1; between 10 and 100 will be between 1 and 2; between 100 and 1000 between 2 and 3. The logarithm of a number less than 1 will be considered later.

It is known that $a^2 \times a^3 = a^5$ and that $a^5 \div a^3 = a^2$. If log. 3 = .4771 and log. 5 = .6990 then $10^{.4771} = 3$ and $10^{.6990} = 5$.

$$5 \times 3 = 10^{.6990} \times 10^{.4771} = 10^{.6990 + .4771} = 10^{1.1761}$$
$$5 \div 3 = 10^{.6990} \div 10^{.4771} = 10^{.6990 - .4771} = 10^{.2219}$$

To multiply two numbers, add their logarithms and this sum is the logarithm of the product; to divide one number by another, subtract the logarithms of the numbers, and the result is the logarithm of the answer.

Example: Multiply 5 × 4 × 20 × 3;

$$\begin{array}{lr} \text{log. } 5 = & .6990 \\ \text{log. } 4 = & .6021 \\ \text{log. } 20 = & 1.3010 \\ \text{log. } 3 = & \underline{.4771} \\ & 3.0792 \end{array}$$

The number corresponding to this logarithm is 1200.

The part of the logarithm to the left of the decimal point *is called the* mantissa. The value of the mantissa determines the location of the decimal point in the answer. Thus

$$\begin{array}{lrl} \text{log.} & 5 = & .6990 \\ \text{log.} & 50 = & 1.6990 \\ \text{log.} & 500 = & 2.6990 \\ \text{log.} & 5000 = & 3.6990 \end{array}$$

It is seen that the mantissa is one unit less than the number of figures to the left of the decimal point.

The part of the logarithm to the right of the decimal point is the same for the same figures irrespective of the location of the decimal point in the number.

Logarithm of a Number Less Than 1.

$$\text{log. } 0.5 = \text{log. } \tfrac{5}{10} = \text{log. } 5 - \text{log. } 10 = .6990 - 1 = 9.6990 - 10$$

$$\text{log. } 0.05 = \text{log. } \tfrac{5}{100} = \text{log. } 5 - \text{log. } 100 = .6990 - 2 = 8.6990 - 10$$

$$\text{log. } 0.005 = \text{log. } \tfrac{5}{1000} = \text{log. } 5 - \text{log. } 1000 = .6990 - 3 = 7.6990 - 10$$

It is seen that the logarithm is followed by − 10 and that the mantissa is 9 if the left-hand figure of the number is in the first decimal place, 8 if in the second, 7 if in the third, etc.

Example: $\dfrac{332. \times 4.13 \times 69.5 \times 0.95 \times 0.00075}{930}$

$$\begin{array}{lrl} \text{log.} & 332 = & 2.5211 \\ \text{log.} & 4.13 = & .6160 \\ \text{log.} & 69.5 = & 1.8420 \\ \text{log.} & 0.95 = & 9.9777 - 10 \\ \text{log.} & 0.00075 = & 6.8751 - 10 \\ \hline & & 21.8319 - 20 \\ \text{log.} & 930 = & 2.9685 \\ \hline & & 18.8634 - 20 \end{array}$$

The number corresponding to .8634 is 730; 18 − 20 is the same as 8 — 10; this indicates that the left-hand figure is in the second decimal place, or 0.0730 = *Ans.*

To raise a number to any power, multiply the log. of the number by the exponent of the power and the result is the log. of the answer.

Examples: $(15)^2$

$$\begin{array}{rl} \log. 15 = & 1.1761 \\ & \underline{2} \\ \log. Ans. = & 2.3522 \\ Ans. = & 225. \end{array}$$

$(1.83)^3$

$$\begin{array}{rl} \log. 1.83 = & .2625 \\ & \underline{3} \\ \log. Ans. = & .7875 \\ Ans. = & 6.13 \end{array}$$

$\sqrt[4]{1.83} = (1.83)^{\frac{1}{4}}$

log. 1.83 = .2625;

multiply by $\frac{1}{4}$ or divide by 4,

$$\begin{array}{rl} \log. Ans. = & .0656 \\ \log. 1.16 = & \underline{.0645} \\ & 11 \end{array}$$

In the table of proportional parts, on page 172, at the right of this line in which .0645 is found, 11 corresponds to 3, so the next figure is 3 and the answer is 1.163.

$\sqrt[5]{0.00373} = (0.00373)^{\frac{1}{5}}$

log. 0.00373 = 7.5717 — 10

this may be written 47.5717 — 50; divide by 5;

$$\begin{array}{rl} \log. Ans. = & 9.5143 - 10; \\ Ans. = & 0.3268 \end{array}$$

TABLES

LOGARITHMS

Nat. Nos.	0	1	2	3	4	5	6	7	8	9	Proportional Parts								
											1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	4	8	12	17	21	25	29	33	37
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	4	8	11	15	19	23	26	30	34
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	3	7	10	14	17	21	24	28	31
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	3	6	10	13	16	19	23	26	29
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	3	6	9	12	15	18	21	24	27
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	3	6	8	11	14	17	20	22	25
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	3	5	8	11	13	16	18	21	24
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	2	5	7	10	12	15	17	20	22
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	2	5	7	9	12	14	16	19	21
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	2	4	7	9	11	13	16	18	20
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	10
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	7
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	7

LOGARITHMS

Nat. Nos.	0	1	2	3	4	5	6	7	8	9	Proportional Parts								
											1	2	3	4	5	6	7	8	9
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	5	6	7
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4	4	5	6	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3	4	4	5	5
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3	4	4	5	5
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	5
82	9138	9143	9149	9154	5159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2	3	3	4	4
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	3	4

AREAS AND CIRCUMFERENCES OF CIRCLES

ADVANCING BY EIGHTHS

Diam.	Area.	Circum.
1-64	.00019	.04909
1-32	.00077	.09818
3-64	.00173	.14726
1-16	.00307	.19635
3-32	.00590	.29452
1-8	.01227	.39270
5-32	.01917	.49087
3-16	.02761	.58905
7-32	.03758	.68722
1-4	.04909	.78540
9-32	.06213	.88357
5-16	.07670	.93175
11-32	.09231	1.0799
3-8	.11045	1.1781
13-32	.12962	1.2763
7-16	.15033	1.3744
15-32	.17257	1.4725
1-2	.19635	1.5708
17-32	.22166	1.6690
9-16	.24850	1.7671
19-32	.27688	1.8653
5-8	.30680	1.9635
21-32	.33824	2.0517
11-16	.37122	2.1598
23-32	.40574	2.2580
3-4	.44179	2.3562
25-32	.47937	2.4544
13-16	.51849	2.5525
27-32	.55914	2.6507
7-8	.60132	2.7489
29-32	.64504	2.8471
15-16	.69029	2.9452
31-32	.73708	3.0434
1.	.7854	3.1416
1-16	.8866	3.3379
1-8	.9940	3.5343
3-16	1.1075	3.7306
1-4	1.2272	3.9270
5-16	1.3530	4.1233
3-8	1.4849	4.3197
7-16	1.6230	4.5160
1-2	1.7671	4.7124
9-16	1.9175	4.9087
5-8	2.0739	5.1051
11-16	2.2365	5.3014
3-4	2.4053	5.4978
13-16	2.5802	5.6941
7-8	2.7612	5.8905
15-16	2.9483	6.0868
2.	3.1416	6.2832
1-16	3.3410	6.4795
1-8	3.5465	6.6759
3-16	3.7583	6.8722
1-4	3.9761	7.0686
5-16	4.2000	7.2649
2 3-8	4.4301	7.4613
7-16	4.6664	7.6576
1-2	4.9087	7.8540
9-16	5.1572	8.0503
5-8	5.4119	8.2467
11-16	5.6727	8.4430
3-4	5.9396	8.6394
13-16	6.2126	8.8357
7-8	6.4918	9.0321
15-16	6.7771	9.2284
3.	7.0686	9.4248
1-16	7.3662	9.6211
1-8	7.6699	9.8175
3-16	7.9798	10.014
1-4	8.2958	10.210
5-16	8.6179	10.407
3-8	8.9462	10.603
7-16	9.2806	10.799
1-2	9.6211	10.996
9-16	9.9678	11.192
5-8	10.321	11.388
11-16	10.680	11.585
3-4	11.045	11.781
13-16	11.416	11.977
7-8	11.793	12.174
15-16	12.177	12.370
4.	12.566	12.566
1-16	12.962	12.763
1-8	13.364	12.959
3-16	13.772	13.155
1-4	14.186	13.352
5-16	14.607	13.548
3-8	15.033	13.744
7-16	15.466	13.941
1-2	15.904	14.137
9-16	16.349	14.334
5-8	16.800	14.530
11-16	17.257	14.726
3-4	17.721	14.923
13-16	18.190	15.119
7-8	18.665	15.315
15-16	19.147	15.512
5.	19.635	15.708
1-16	20.129	15.904
1-8	20.629	16.101
3-16	21.135	16.297
1-4	21.648	16.493
5-16	22.166	16.690
3-8	22.691	16.886
7-16	23.221	17.082
1-2	23.758	17.279
9-16	24.301	17.475
5-8	24.850	17.671
11-16	25.406	17.868
3-4	25.967	18.064
13-16	26.535	18.261
7-8	27.109	18.457
15-16	27.688	18.653
6.	28.274	18.850
1-8	29.465	19.242
1-4	30.680	19.635
3-8	31.919	20.028
1-2	33.183	20.420
5-8	34.472	20.813
3-4	35.785	21.206
7-8	37.122	21.598
7.	38.485	21.991
1-8	39.871	22.384
1-4	41.282	22.776
3-8	42.718	23.169
1-2	44.179	23.562
5-8	45.664	23.955
3-4	47.173	24.347
7-8	48.707	24.740
8.	50.265	25.133
1-8	51.849	25.525
1-4	53.456	25.918
3-8	55.088	26.311
1-2	56.745	26.704
5-8	58.426	27.096
3-4	60.132	27.489
7-8	61.862	27.882
9.	63.617	28.274
1-8	65.397	28.667
1-4	67.201	29.060
3-8	69.029	29.452
1-2	70.882	29.845
5-8	72.760	30.238
3-4	74.662	30.631
7-8	76.589	31.023
10.	78.540	31.416
1-8	80.516	31.809
1-4	82.516	32.201
3-8	84.541	32.594
1-2	86.590	32.987
5-8	88.664	33.379
3-4	90.763	33.772
7-8	92.886	34.165
11.	95.033	34.558
1-8	97.205	34.950
1-4	99.402	35.343
3-8	101.62	35.736
1-2	103.87	36.128
5-8	106.14	36.521
3-4	108.43	36.914
7-8	110.75	37.306
12.	113.10	37.699
1-8	115.47	38.092
1-4	117.86	38.485
3-8	120.28	38.877
1-2	122.72	39.270
5-8	125.19	39.663
3-4	127.68	40.055
7-8	130.19	40.448

AREAS AND CIRCUMFERENCES OF CIRCLES

For Diameters from $\frac{1}{10}$ to 99, advancing by Tenths

Diam.	Area.	Circum.	Diam.	Area.	Circum.	Diam.	Area.	Circum.
0.0			5.0	19.6350	15.7080	10.0	78.5398	31.4159
.1	.007854	.31416	.1	20.4282	16.0221	.1	80.1185	31.7301
.2	.031416	.62832	.2	21.2372	16.3363	.2	81.7128	32.0442
.3	.070686	.94248	.3	22.0618	16.6504	.3	83.3229	32.3584
.4	.12566	1.2566	.4	22.9022	16.9646	.4	84.9487	32.6726
.5	.19635	1.5708	.5	23.7583	17.2788	.5	86.5901	32.9867
.6	.28274	1.8850	.6	24.6301	17.5929	.6	88.2473	33.3009
.7	.38485	2.1991	.7	25.5176	17.9071	.7	89.9202	33.6150
.8	.50266	2.5133	.8	26.4208	18.2212	.8	91.6088	33.9292
.9	.63617	2.8274	.9	27.3397	18.5354	.9	93.3132	34.2434
1.0	.7854	3.1416	6.0	28.2743	18.8496	11.0	95.0332	34.5575
.1	.9503	3.4558	.1	29.2247	19.1637	.1	96.7689	34.8717
.2	1.1310	3.7699	.2	30.1907	19.4779	.2	98.5203	35.1858
.3	1.3273	4.0841	.3	31.1725	19.7920	.3	100.2875	35.5000
.4	1.5394	4.3982	.4	32.1699	20.1062	.4	102.0703	35.8142
.5	1.7671	4.7124	.5	33.1831	20.4204	.5	103.8689	36.1283
.6	2.0106	5.0265	.6	34.2119	20.7345	.6	105.6832	36.4425
.7	2.2698	5.3407	.7	35.2565	21.0487	.7	107.5132	36.7566
.8	2.5447	5.6549	.8	36.3168	21.3628	.8	109.3588	37.0708
.9	2.8353	5.9690	.9	37.3928	21.6770	.9	111.2202	37.3850
2.0	3.1416	6.2832	7.0	38.4845	21.9911	12.0	113.0973	37.6991
.1	3.4636	6.5973	.1	39.5919	22.3053	.1	114.9901	38.0133
.2	3.8013	6.9115	.2	40.7150	22.6195	.2	116.8987	38.3274
.3	4.1548	7.2257	.3	41.8539	22.9336	.3	118.8229	38.6416
.4	4.5239	7.5398	.4	43.0084	23.2478	.4	120.7628	38.9557
.5	4.9087	7.8540	.5	44.1786	23.5619	.5	122.7185	39.2699
.6	5.3093	8.1681	,6	45.3646	23.8761	.6	124.6898	39.5841
.7	5.7256	8.4823	.7	46.5663	24.1903	.7	126.6769	39.8982
.8	6.1575	8.7965	.8	47.7836	24.5044	.8	128.6796	40.2124
.9	6.6052	9.1106	.9	49.0167	24.8186	.9	130.6981	40.5265
3.0	7.0686	9.4248	8.0	50.2655	25.1327	13.0	132.7323	40.8407
.1	7.5477	9.7389	.1	51.5300	25.4469	.1	134.7822	41.1549
.2	8.0425	10.0531	.2	52.8102	25.7611	.2	136.8478	41.4690
.3	8.5530	10.3673	.3	54.1061	26.0752	.3	138.9291	41.7832
.4	9.0792	10.6814	.4	55.4177	26.3894	.4	141.0261	42.0973
.5	9.6211	10.9956	.5	56.7450	26.7035	.5	143.1388	42.4115
.6	10.1788	11.3097	.6	58.0880	27.0177	.6	145.2672	42.7257
.7	10.7521	11.6239	.7	59.4468	27.3319	.7	147.4114	42.0398
.8	11.3411	11.9381	.8	60.8212	27.6460	.8	149.5712	43.3540
.9	11.9459	12.2522	.9	62.2114	27.9602	.9	151.7468	43.6681
4.0	12.5664	12.5664	9.0	63.6173	28.2743	14.0	153.9380	43.9823
.1	13.2025	12.8805	.1	65.0388	28.5885	.1	156.1450	44.2965
.2	13.8544	13.1947	.2	66.4761	28.9027	.2	158.3677	44.6106
.3	14.5220	13.5088	.3	67.9291	29.2168	.3	160.6061	44.9248
.4	15.2053	13.8230	.4	69.3978	29.5310	.4	162.8602	45.2389
.5	15.9043	14.1372	.5	70.8822	29.8451	.5	165.1300	45.5531
.6	16.6190	14.4513	.6	72.3823	30.1593	.6	167.4155	45.8673
.7	17.3494	14.7655	.7	73.8981	30.4734	.7	169.7167	46.1814
.8	18.0956	15.0796	.8	75.4296	30.7876	.8	172.0336	46.4956
.9	18.8574	15.3938	.9	76.9769	31.1018	.9	174.3662	46.8097

AREAS AND CIRCUMFERENCES OF CIRCLES

For Diameters from $\frac{1}{10}$ to 99, advancing by Tenths

Diam.	Area.	Circum.	Diam.	Area.	Circum.	Diam.	Area.	Circum.
15.0	176.7146	47.1239	20.0	314.1593	62.8319	25.0	490.8739	78.5398
.1	179.0786	47.4380	.1	317.3087	63.1460	.1	494.8087	78.8540
.2	181.4584	47.7522	.2	320.4739	63.4602	.2	498.7592	79.1681
.3	183.8539	48.0664	.3	323.6547	63.7743	.3	502.7255	79.4823
.4	186.2650	48.3805	.4	326.8513	64.0885	.4	506.7075	79.7965
.5	188.6919	48.6947	.5	330.0636	64.4026	.5	510.7052	80.1106
.6	191.1345	49.0088	.6	333.2916	64.7168	.6	514.7185	80.4248
.7	193.5928	49.3230	.7	336.5353	65.0310	.7	518.7476	80.7389
.8	196.0668	49.6372	.8	339.7947	65.3451	.8	522.7924	81.0531
.9	198.5565	49.9513	.9	343.0698	65.6593	.9	526.8529	81.3672
16.0	201.0619	50.2655	21.0	346.3606	65.9734	26.0	530.9292	81.6814
.1	203.5831	50.5796	.1	349.6671	66.2876	.1	535.0211	81.9956
.2	206.1199	50.8938	.2	352.9894	66.6018	.2	539.1287	82.3097
.3	208.6724	51.2080	.3	356.3273	66.9159	.3	543.2521	82.6239
.4	211.2407	51.5221	.4	359.6809	67.2301	.4	547.3911	82.9380
.5	213.8246	51.8363	.5	363.0503	67.5442	.5	551.5459	83.2522
.6	216.4243	52.1504	.6	366.4354	67.8584	.6	555.7163	83.5664
.7	219.0397	52.4646	.7	369.8361	68.1726	.7	559.9025	83.8805
.8	221.6708	52.7788	.8	373.2526	68.4867	.8	564.1044	84.1947
.9	224.3176	53.0929	.9	376.6848	68.8009	.9	568.3220	84.5088
17.0	226.9801	53.4071	22.0	380.1327	69.1150	27.0	572.5553	84.8230
.1	229.6583	53.7212	.1	383.5963	69.4292	.1	576.8043	85.1372
.2	232.3522	54.0354	.2	387.0756	69.7434	.2	581.0690	85.4513
.3	235.0618	54.3496	.3	390.5707	70.0575	.3	585.3494	85.7655
.4	237.7871	54.6637	.4	394.0814	70.3717	.4	589.6455	86.0796
.5	240.5282	54.9779	.5	397.6078	70.6858	.5	593.9574	86.3938
.6	243.2849	55.2920	.6	401.1500	71.0000	.6	598.2849	86.7080
.7	246.0574	55.6062	.7	404.7078	71.3142	.7	602.6282	87.0221
.8	248.8456	55.9203	.8	408.2814	71.6283	.8	606.9871	87.3363
.9	251.6494	56.2345	.9	411.8707	71.9425	.9	611.3618	87.6504
18.0	254.4690	56.5486	23.0	415.4756	72.2566	28.0	615.7522	87.9646
.1	257.3043	56.8628	.1	419.0963	72.5708	.1	620.1582	88.2788
.2	260.1553	57.1770	.2	422.7327	72.8849	.2	624.5800	88.5929
.3	263.0220	57.4911	.3	426.3848	73.1991	.3	629.0175	88.9071
.4	265.9044	57.8053	.4	430.0526	73.5133	.4	633.4707	89.2212
.5	268.8025	58.1195	.5	433.7361	73.8274	.5	637.9397	89.5354
.6	271.7164	58.4336	.6	437.4354	74.1416	.6	642.4243	89.8495
.7	274.6459	58.7478	.7	441.1503	74.4557	.7	646.9246	90.1637
.8	277.5911	59.0619	.8	444.8809	74.7699	.8	651.4407	90.4779
.9	280.5521	59.3761	.9	448.6273	75.0841	.9	655.9724	90.7920
19.0	283.5287	59.6903	24.0	452.3893	75.3982	29.0	660.5199	91.1062
.1	286.5211	60.0044	.1	456.1671	75.7124	.1	665.0830	91.4203
.2	289.5292	60.3186	.2	459.9606	76.0265	.2	669.6619	91.7345
.3	292.5530	60.6327	.3	463.7698	76.3407	.3	674.2565	92.0487
.4	295.5925	60.9469	.4	467.5947	76.6549	.4	678.8668	92.3628
.5	298.6477	61.2611	.5	471.4352	76.9690	.5	683.4928	92.6770
.6	301.7186	61.5752	.6	475.2916	77.2832	.6	688.1345	92.9911
.7	304.8052	61.8894	.7	479.1636	77.5973	.7	692.7919	93.3053
.8	307.9075	62.2035	.8	483.0513	77.9115	.8	697.4650	93.6195
.9	311.0255	62.5177	.9	486.9547	78.2257	.9	702.1538	93.9336

AREAS AND CIRCUMFERENCES OF CIRCLES

For Diameters from $\frac{1}{10}$ to 99, advancing by Tenths

Diam.	Area.	Circum.	Diam.	Area.	Circum.	Diam.	Area.	Circum.
30.0	706.8583	94.2478	35.0	962.1128	109.9557	40.0	1256.6371	125.6637
.1	711.5786	94.5619	.1	967.6184	110.2699	.1	1262.9281	125.9779
.2	716.3145	94.8761	.2	973.1397	110.5841	.2	1269.2348	126.2920
.3	721.0662	95.1903	.3	978.6768	110.8982	.3	1275.5573	126.6062
.4	725.8336	95.5044	.4	984.2296	111.2124	.4	1281.8955	126.9203
.5	730.6167	95.8186	.5	989.7980	111.5265	.5	1288.2493	127.2345
.6	735.4154	96.1327	.6	995.3822	111.8407	.6	1294.6189	127.5487
.7	740.2299	96.4469	.7	1000.9821	112.1549	.7	1301.0042	127.8628
.8	745.0601	96.7611	.8	1006.5977	112.4690	.8	1307.4052	128.1770
.9	749.9060	97.0752	.9	1012.2290	112.7832	.9	1313.8219	128.4911
31.0	754.7676	97.3894	36.0	1017.8760	113.0973	41.0	1320.2543	128.8053
.1	759.6450	97.7035	.1	1023.5387	113.4115	.1	1326.7024	129.1195
.2	764.5380	98.0177	.2	1029.2172	113.7257	.2	1333.1663	129.4336
.3	769.4467	98.3319	.3	1034.9113	114.0398	.3	1339.6458	129.7478
.4	774.3712	98.6460	.4	1040.6212	114.3540	.4	1346.1410	130.0619
.5	779.3113	98.9602	.5	1046.3467	114.6681	.5	1352.6520	130.3761
.6	784.2672	99.2743	.6	1052.0880	114.9823	.6	1359.1786	130.6903
.7	789.2388	99.5885	.7	1057.8449	115.2965	.7	1365.7210	131.0044
.8	794.2260	99.9026	.8	1063.6176	115.6106	.8	1372.2791	131.3186
.9	799.2290	100.2168	.9	1069.4060	115.9248	.9	1378.8529	131.6327
32.0	804.2477	100.5310	37.0	1075.2101	116.2389	42.0	1385.4424	131.9469
.1	809.2821	100.8451	.1	1081.0299	116.5531	.1	1392.0476	132.2611
.2	814.3322	101.1593	.2	1086.8654	116.8672	.2	1398.6685	132.5752
.3	819.3980	101.4734	.3	1092.7166	117.1814	.3	1405.3051	132.8894
.4	824.4796	101.7876	.4	1098.5835	117.4956	.4	1411.9574	133.2035
.5	829.5768	102.1018	.5	1104.4662	117.8097	.5	1418.6254	133.5177
.6	834.6898	102.4159	.6	1110.3645	118.1239	.6	1425.3092	133.8318
.7	839.8185	102.7301	.7	1116.2786	118.4380	.7	1432.0086	134.1460
.8	844.9628	103.0442	.8	1122.2083	118.7522	.8	1438.7238	134.4602
.9	850.1229	103.3584	.9	1128.1538	119.0664	.9	1445.4546	134.7743
33.0	855.2986	103.6726	38.0	1134.1149	119.3805	43.0	1452.2012	135.0885
.1	860.4902	103.9867	.1	1140.0918	119.6947	.1	1458.9635	135.4026
.2	865.6973	104.3009	.2	1146.0844	120.0088	.2	1465.7415	135.7168
.3	870.9202	104.6150	.3	1152.0927	120.3230	.3	1472.5352	136.0310
.4	876.1588	104.9292	.4	1158.1167	120.6372	.4	1479.3446	136.3451
.5	881.4131	105.2434	.5	1164.1564	120.9513	.5	1486.1697	136.6593
.6	886.6831	105.5575	.6	1170.2118	121.2655	.6	1493.0105	136.9734
.7	891.9688	105.8717	.7	1176.2830	121.5796	.7	1499.8670	137.2876
.8	897.2703	106.1858	.8	1182.3698	121.8938	.8	1506.7393	137.6018
.9	902.5874	106.5000	.9	1188.4724	122.2080	.9	1513.6272	137.9159
34.0	907.9203	106.8142	39.0	1194.5906	122.5221	44.0	1520.5308	138.2301
.1	913.2688	107.1283	.1	1200.7246	122.8363	.1	1527.4502	138.5442
.2	918.6331	107.4425	.2	1206.8742	123.1504	.2	1534.3853	138.8584
.3	924.0131	107.7566	.3	1213.0396	123.4646	.3	1541.3360	139.1726
.4	929.4088	108.0708	.4	1219.2207	123.7788	.4	1548.3025	139.4867
.5	934.8202	108.3849	.5	1225.4175	124.0929	.5	1555.2847	139.8009
.6	940.2473	108.6991	.6	1231.6300	124.4071	.6	1562.2826	140.1153
.7	945.6901	109.0133	.7	1237.8582	124.7212	.7	1569.2962	140.4292
.8	951.1486	109.3274	.8	1244.1021	125.0354	.8	1576.3255	140.7434
.9	956.6228	109.6416	.9	1250.3617	125.3495	.9	1583.3706	141.0575

AREAS AND CIRCUMFERENCES OF CIRCLES

For Diameters from $\frac{1}{10}$ to 99, advancing by Tenths

Diam.	Area.	Circum.	Diam.	Area.	Circum.	Diam.	Area.	Circum.
45.0	1590.4313	141.3717	50.0	1963.4954	157.0796	55.0	2375.8294	172.7876
.1	1597.5077	141.6858	.1	1971.3572	157.3938	.1	2384.4767	173.1017
.2	1604.5999	142.0000	.2	1979.2348	157.7080	.2	2393.1396	173.4159
.3	1611.7077	142.3142	.3	1987.1280	158.0221	.3	2401.8183	173.7301
.4	1618.8313	142.6283	.4	1995.0370	158.3363	.4	2410.5126	174.0442
.5	1625.9705	142.9425	.5	2002.9617	158.6504	.5	2419.2227	174.3584
.6	1633.1255	143.2566	.6	2010.9020	158.9646	.6	2427.9485	174.6726
.7	1640.2962	143.5708	.7	2018.8581	159.2787	.7	2436.6899	174.9867
.8	1647.4826	143.8849	.8	2026.8299	159.5929	.8	2445.4471	175.3009
.9	1654.6847	144.1991	.9	2034.8174	159.9071	.9	2454.2200	175.6150
46.0	1661.9025	144.5133	51.0	2042.8206	160.2212	56.0	2463.0086	175.9292
.1	1669.1360	144.8274	.1	2050.8395	160.5354	.1	2471.8130	176.2433
.2	1676.3853	145.1416	.2	2058.8742	160.8495	.2	2480.6330	176.5575
.3	1683.6502	145.4557	.3	2066.9245	161.1637	.3	2489.4687	176.8717
.4	1690.9303	145.7699	.4	2074.9905	161.4779	.4	2498.3201	177.1858
.5	1698.2272	146.0841	.5	2083.0723	161.7920	.5	2507.1873	177.5000
.6	1705.5392	145.3982	.6	2091.1697	162.1062	.6	2516.0701	177.8141
.7	1712.8670	145.7124	.7	2099.2829	162.4203	.7	2524.9687	178.1283
.8	1720.2105	147.0265	.8	2107.4118	162.7345	.8	2533.8830	178.4425
.9	1727.5697	147.3407	.9	2115.5563	163.0487	.9	2542.8129	178.7566
47.0	1734.9445	147.6550	52.0	2123.7166	163.3628	57.0	2551.7586	179.0708
.1	1742.3351	147.9690	.1	2131.8926	163.6770	.1	2560.7200	179.3849
.2	1749.7414	148.2832	.2	2140.0843	163.9911	.2	2569.6971	179.6991
.3	1757.1635	148.5973	.3	2148.2917	164.3053	.3	2578.6899	180.0133
.4	1764.6012	148.9115	.4	2156.5149	164.6195	.4	2587.6985	180.3274
.5	1772.0546	149.2257	.5	2164.7537	164.9336	.5	2596.7227	180.6416
.6	1779.5237	149.5393	.6	2173.0082	165.2479	.6	2605.7626	180.9557
.7	1787.0086	149.8540	.7	2181.2785	165.5619	.7	2614.8183	181.2699
.8	1794.5091	150.1681	.8	2189.5644	165.8761	.8	2623.8896	181.5841
.9	1802.0254	150.4823	.9	2197.8661	166.1903	.9	2632.9767	181.8982
48.0	1809.5574	150.7964	53.0	2206.1834	166.5044	58.0	2642.0794	182.2124
.1	1817.1050	151.1106	.1	2214.5165	166.8186	.1	2651.1979	182.5265
.2	1824.6684	151.4248	.2	2222.8653	167.1327	.2	2660.3321	182.8407
.3	1832.2475	151.7389	.3	2231.2298	167.4469	.3	2669.4820	183.1549
.4	1839.8423	152.0531	.4	2239.6100	167.7610	.4	2678.6476	183.4690
.5	1847.4528	152.3672	.5	2248.0059	168.0752	.5	2687.8289	183.7832
.6	1855.0790	152.6814	.6	2256.4175	168.3894	.6	2697.0259	184.0973
.7	1862.7210	152.9956	.7	2264.8448	168.7035	.7	2706.2386	184.4115
.8	1870.3786	153.3097	.8	2273.2879	169.0177	.8	2715.4670	184.7256
.9	1878.0519	153.6239	.9	2281.7466	169.3318	.9	2724.7112	185.0398
49.0	1885.7409	153.9380	54.0	2290.2210	169.6460	59.0	2733.9710	185.3540
.1	1893.4457	154.2522	.1	2298.7112	169.9602	.1	2743.2466	185.6681
.2	1901.1662	154.5664	.2	2307.2171	170.2743	.2	2752.5378	185.9823
.3	1908.9024	154.8805	.3	2315.7386	170.5885	.3	2761.8448	186.2964
.4	1916.6543	155.1947	.4	2324.2759	170.9026	.4	2771.1675	186.6106
.5	1924.4218	155.5088	.5	2332.8289	171.2168	.5	2780.5058	186.9248
.6	1932.2051	155.8230	.6	2341.3976	171.5310	.6	2789.8599	187.2389
.7	1940.0042	156.1372	.7	2349.9820	171.8451	.7	2799.2297	187.5531
.8	1947.8189	156.4513	.8	2358.5821	172.1593	.8	2808.6152	187.8672
.9	1955.6493	156.7655	.9	2367.1979	172.4735	.9	2818.0165	188.1814

AREAS AND CIRCUMFERENCES OF CIRCLES

For Diameters from $\frac{1}{10}$ to 99, advancing by Tenths

Diam.	Area.	Circum.	Diam.	Area.	Circum.	Diam.	Area.	Circum.
60.0	2827.4334	188.4956	65.0	3318.3072	204.2035	70.0	3848.4510	219.9115
.1	2836.8660	188.8097	.1	3328.5253	204.5176	.1	3859.4544	220.2256
.2	2846.3144	189.1239	.2	3338.7590	204.8318	.2	3870.4736	220.5398
.3	2855.7784	189.4380	.3	3349.0085	205.1460	.3	3881.5084	220.8540
.4	2865.2582	189.7522	.4	3359.2736	205.4602	.4	3892.5590	221.1681
.5	2874.7536	190.0664	.5	3369.5545	205.7743	.5	3903.6252	221.4823
.6	2884.2648	190.3805	.6	3379.8510	206.0885	.6	3914.7072	221.7964
.7	2893.7917	190.6947	.7	3390.1633	206.4026	.7	3925.8049	222.1106
.8	2903.3343	191.0088	.8	3400.4913	206.7168	.8	3936.9182	222,4248
.9	2912.8926	191.3230	.9	3410.8350	207.0310	.9	3948.0473	222.7389
61.0	2922.4666	191.6372	66.0	3421.1944	207.3451	71.0	3959.1921	223.0531
.1	2932.0563	191.9513	.1	3431.5695	207.6593	.1	3970.3526	223.3672
.2	2941.6617	192.2655	.2	3441.9603	207.9734	.2	3981.5289	223.6814
.3	2951.2828	192.5796	.3	3452.3669	208.2876	.3	3992.7208	223.9956
.4	2960.9197	192.8938	.4	3462.7891	208.6017	.4	4003.9284	224.3097
.5	2970.5722	193.2079	.5	3473.2270	208.9159	.5	4015.1518	224.6239
.6	2980.2405	193.5221	.6	3483.6807	209.2301	.6	4026.3908	224.9380
.7	2989.9244	193.8363	.7	3494.1500	209.5442	.7	4037.6456	225.2522
.8	2999.6241	194.1504	.8	3504.6351	209.8584	.8	4048.9160	225.5664
.9	3009.3395	194.4646	.9	3515.1359	210.1725	.9	4060.2022	225.8805
62.0	3019.0705	194.7787	67.0	3525.6524	210.4867	72.0	4071.5041	226.1947
.1	3028.8173	195.0929	.1	3536.1845	210.8009	.1	4082.8217	226.5088
.2	3038.5798	195.4071	.2	3546.7324	211.1150	.2	4094.1550	226.8230
.3	3048.3580	195.7212	.3	3557.2960	211.4292	.3	4105.5040	227.1371
.4	3058.1520	196.0354	.4	3567.8754	211.7433	.4	4116.8687	227.4513
.5	3067.9616	196.3495	.5	3578.4704	212.0575	.5	4128.2491	227.7655
.6	3077.7869	196.6637	.6	3589.0811	212.3717	.6	4139.6452	228.0796
.7	3087.6279	196.9779	.7	3599.7075	212.6858	.7	4151.0571	228.3938
.8	3097.4847	197.2920	.8	3610.3497	213.0000	.8	4162.4846	228.7079
.9	3107.3571	197.6062	.9	3621.0075	213.3141	.9	4173.9279	229.0221
63.0	3117.2453	197.9203	68.0	3631.6811	213.6283	73.0	4185.3868	229.3363
.1	3127.1492	198.2345	.1	3642.3704	213.9425	.1	4196.8615	229.6504
.2	3137.0688	198.5487	.2	3653.0754	214.2566	.2	4208.3519	229.9646
.3	3147.0040	198.8628	.3	3663.7960	214.5708	.3	4219.8579	230.2787
.4	3156.9550	199.1770	.4	3674.5324	214.8849	.4	4231.3797	230.5929
.5	3166.9217	199.4911	.5	3685.2845	215.1991	.5	4242.9172	230.9071
.6	3176.9043	199.8053	.6	3696.0523	215.5133	.6	4254.4704	231.2212
.7	3186.9023	200.1195	.7	3706.8359	215.8274	.7	4266.0394	231.5354
.8	3196.9161	200.4336	.8	3717.6351	216.1416	.8	4277.6240	231.8495
.9	3206.9456	200.7478	.9	3728.4500	216.4556	.9	4289.2243	232.1637
64.0	3216.9909	201.0620	69.0	3739.2807	216.7699	74.0	4300.8403	232.4779
.1	3227.0518	201.3761	.1	3750.1270	217.0841	.1	4312.4721	232.7920
.2	3237.1285	201.6902	.2	3760.9891	217.3982	.2	4324.1195	233.1062
.3	3247.2222	202.0044	.3	3771.8668	217.7124	.3	4335.7827	233.4203
.4	3257.3289	202.3186	.4	3782.7603	218.0265	.4	4347.4616	233.7345
.5	3267.4527	202.6327	.5	3793.6695	218.3407	.5	4359.1562	234.0487
.6	3277.5922	202.9469	.6	3804.5944	218.6548	.6	4370.8664	234.3628
.7	3287.7474	203.2610	.7	3815.5350	218.9690	.7	4382.5924	234.6770
.8	3297.9183	203.5752	.8	3826.4913	219.2832	.8	4394.3341	234.9911
.9	3308.1049	203.8894	.9	3837.4633	219.5973	.9	4406.0916	235.3053

AREAS AND CIRCUMFERENCES OF CIRCLES

For Diameters from $\frac{1}{10}$ to 99, advancing by Tenths

Diam.	Area.	Circum.	Diam.	Area.	Circum.	Diam.	Area.	Circum.
75.0	4417.8647	235.6194	80.0	5026.5482	251.3274	85.0	5674.5017	267.0354
.1	4429.6535	235.9336	.1	5039.1225	251.6416	.1	5687.8614	267.3495
.2	4441.4580	236.2478	.2	5051.7124	251.9557	.2	5701.2367	267.6637
.3	4453.2783	236.5619	.3	5064.3180	252.2699	.3	5714.6277	267.9779
.4	4465.1142	236.8761	.4	5076.9394	252.5840	.4	5728.0345	268.2920
.5	4476.9659	237.1902	.5	5089.5764	252.8982	.5	5741.4569	268.6062
.6	4488.8332	237.5044	.6	5102.2292	253.2124	.6	5754.8951	268.9203
.7	4500.7163	237.8186	.7	5114.8977	253.5265	.7	5768.3490	269.2345
.8	4512.6151	238.1327	.8	5127.5819	253.8407	.8	5781.8185	269.5486
.9	4524.5296	238.4469	.9	5140.2818	254.1548	.9	5795.3038	269.8628
76.0	4536.4598	238.7610	81.0	5152.9973	254.4690	86.0	5808.8048	270.1770
.1	4548.4057	239.0752	.1	5165.7287	254.7832	.1	5822.3215	270.4911
.2	4560.3673	239.3894	.2	5178.4757	255.0973	.2	5835.8539	270.8053
.3	4572.3446	239.7035	.3	5191.2384	255.4115	.3	5849.4020	271.1194
.4	4584.3377	240.0177	.4	5204.0168	255.7256	.4	5862.9659	271.4336
.5	4596.3464	240.3318	.5	5216.8110	256.0398	.5	5876.5454	271.7478
.6	4608.3708	240.6460	.6	5229.6208	256.3540	.6	5890.1407	272.0619
.7	4620.4110	240.9602	.7	5242.4463	256.6681	.7	5903.7516	272.3761
.8	4632.4669	241.2743	.8	5255.2876	256.9823	.8	5917.3783	272.6902
.9	4644.5384	241.5885	.9	5268.1446	257.2966	.9	5931.0206	273.0044
77.0	4656.6257	241.9026	82.0	5281.0173	257.6106	87.0	5944.6787	273.3186
.1	4668.7287	242.2168	.1	5293.9056	257.9247	.1	5958.3525	273.6327
.2	4680.8474	242.5310	.2	5306.8097	258.2389	.2	5972.0420	273.9469
.3	4692.9818	242.8451	.3	5319.7295	258.5531	.3	5985.7472	274.2610
.4	4705.1319	243.1592	.4	5332.6650	258.8672	.4	5999.4681	274.5752
.5	4717.2977	243.4734	.5	5345.6162	259.1814	.5	6013.2047	274.8894
.6	4729.4792	243.7876	.6	5358.5832	259.4956	.6	6026.9570	275.2035
.7	4741.6765	244.1017	.7	5371.5658	259.8097	.7	6040.7250	275.5177
.8	4753.8894	244.4159	.8	5384.5641	260.1239	.8	6054.5088	275.8318
.9	4766.1181	244.7301	.9	5397.5782	260.4380	.9	6068.3082	276.1460
78.0	4778.3624	245.0442	83.0	5410.6079	260.7522	88.0	6082.1234	276.4602
.1	4790.6225	245.3584	.1	5423.6534	261.0663	.1	6095.9542	276.7743
.2	4802.8983	245.6725	.2	5436.7146	261.3805	.2	6109.8008	277.0885
.3	4815.1897	245.9867	.3	5449.7915	261.6947	.3	6123.6631	277.4026
.4	4827.4969	246.3009	.4	5462.8840	262.0088	.4	6137.5411	277.7168
.5	4839.8198	246.6150	.5	5475.9923	262.3230	.5	6151.4348	278.0309
.6	4852.1584	246.9292	.6	5489.1163	262.6371	.6	6165.3442	278.3451
.7	4864.5128	247.2433	.7	5502.2561	262.9513	.7	6179.2693	278.6593
.8	4876.8828	247.5575	.8	5515.4115	263.2655	.8	6193.2101	278.9740
.9	4889.2685	247.8717	.9	5528.5826	263.5796	.9	6207.1666	279.2876
79.0	4901.6699	248.1858	84.0	5541.7694	263.8938	89.0	6221.1389	279.6017
.1	4914.0871	248.5000	.1	5554.9720	264.2079	.1	6235.1268	279.9159
.2	4926.5199	248.8141	.2	5568.1902	264.5221	.2	6249.1304	280.2301
.3	4938.9685	249.1283	.3	5581.4242	264.8363	.3	6263.1498	280.5442
.4	4951.4328	249.4425	.4	5594.6739	265.1514	.4	6277.1849	280.8584
.5	4963.9127	249.7566	.5	5607.9392	265.4646	.5	6291.2356	281.1725
.6	4976.4084	250.0708	.6	5621.2203	265.7787	.6	6305.3021	281.4867
.7	4988.9198	250.3850	.7	5634.5171	266.0929	.7	6319.3843	281.8009
.8	5001.4469	250.6991	.8	5647.8296	266.4071	.8	6333.4822	282.1150
.9	*5013.9897*	251.0133	.9	5661.1578	266.7212	.9	6347.5958	282.4292

AREAS AND CIRCUMFERENCES OF CIRCLES

For Diameters from $\frac{1}{10}$ to 99, advancing by Tenths

Diam.	Area.	Circum.	Diam.	Area.	Circum.	Diam.	Area.	Circum.
90.0	6361.7251	282.7433	93.0	6792.9087	292.1681	96.0	7238.2295	301.5929
.1	6375.8701	283.0575	.1	6807.5250	292.4823	.1	7253.3170	301.9071
.2	6390.0309	283.3717	.2	6822.1569	292.7964	.2	7268.4202	302.2212
.3	6404.2073	283.6858	.3	6836.8046	293.1106	.3	7283.5391	302.5354
.4	6418.3995	284.0000	.4	6851.4680	293.4248	.4	7298.6737	302.8405
.5	6432.6073	284.3141	.5	6866.1471	293.7389	.5	7313.8240	303.1637
.6	6446.8309	284.6283	.6	6880.8419	294.0531	.6	7328.9901	303.4779
.7	6461.0701	284.9425	.7	6895.5524	294.3672	.7	7344.1718	303.7920
.8	6475.3251	285.2566	.8	6910.2786	294.6814	.8	7359.3693	304.1062
.9	6489.5958	285.5708	.9	6925.0205	294.9956	.9	7374.5824	304.4203
91.0	6503.8322	285.8849	94.0	6939.7782	295.3097	97.0	7389.8113	304.7345
.1	6518.1843	286.1991	.1	6954.5515	295.6239	.1	7405.0559	305.0486
.2	6532.5021	286.5133	.2	6969.3106	295.9380	.2	7420.3162	305.3628
.3	6546.8356	286.8274	.3	6984.1453	296.2522	.3	7435.5922	305.6770
.4	6561.1848	287.1416	.4	6998.9658	296.5663	.4	7450.8839	305.9911
.5	6575.5498	287.4557	.5	7013.8019	296.8805	.5	7466.1913	306.3053
.6	6589.9304	287.7699	.6	7028.6538	297.1947	.6	7481.5144	306.6194
.7	6604.3268	288.0840	.7	7043.5214	297.5088	.7	7496.8532	306.9336
.8	6618.7388	288.3982	.8	7058.4047	297.8230	.8	7512.2078	307.2478
.9	6633.1666	288.7124	.9	7073.3033	298.1371	.9	7527.5780	307.5619
92.0	6647.6101	289.0265	95.0	7088.2184	298.4513	98.0	7542.9640	307.8761
.1	6662.0692	289.3407	.1	7103.1488	298.7655	.1	7558.3656	308.1902
.2	6676.5441	289.6548	.2	7118.1950	299.0796	.2	7573.7830	308.5044
.3	6691.0347	289.9690	.3	7133.0568	299.3938	.3	7589.2161	308.8186
.4	6705.5410	290.2832	.4	7148.0343	299.7079	.4	7604.6648	309.1327
.5	6720.0630	290.5973	.5	7163.0276	300.0221	.5	7620.1293	309.4469
.6	6734.6008	290.9115	.6	7178.0366	300.3363	.6	7635.6095	309.7610
.7	6749.1542	291.2256	.7	7193.0612	300.6504	.7	7651.1054	310.0752
.8	6763.7233	291.5398	.8	7208.1016	300.9646	.8	7666.6170	310.3894
.9	6778.3082	291.8540	.9	7223.1577	301.2787	.9	7682.1444	310.7035

DECIMAL EQUIVALENTS OF FRACTIONS OF ONE INCH

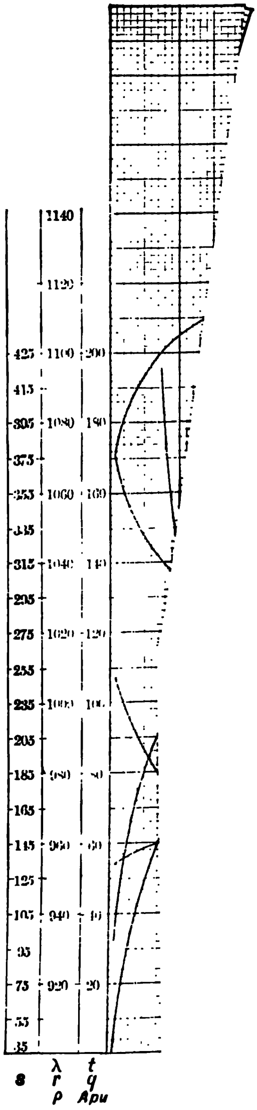
1-64	.015625	17-64	.265625	33-64	.515625	49-64	.765625
1-32	.03125	9-32	.28125	17-32	.53125	25-32	.78125
3-64	.046875	19-64	.296875	35-64	.546875	51-64	.796875
1-16	.0625	**5-16**	.3125	**9-16**	.5625	**13-16**	.8125
5-64	.078125	21-64	.328125	37-64	.578125	53-64	.828125
3-32	.09375	11-32	.34375	19-32	.59375	27-32	.84375
7-64	.109375	23-64	.359375	39-64	.609375	55-64	.859375
1-8	.125	**3-8**	.375	**5-8**	.625	**7-8**	.875
9-64	.140625	25-64	.390625	41-64	.640625	57-64	.890625
5-32	.15625	13-32	.40625	21-32	.65625	29-32	.90625
11-64	.171875	27-64	.421875	43-64	.671875	59-64	.921875
3-16	.1875	**7-16**	.4375	**11-16**	.6875	**15-16**	.9375
13-64	.203125	29-64	.453125	45-64	.703125	61-64	.953125
7-32	.21875	15-32	.46875	23-32	.71875	31-32	.96875
15-64	.234375	31-64	.484375	47-64	.734375	63-64	.984375
1-4	.25	**1-2**	.50	**3-4**	.75	**1**	1.

WEIGHT OF A CUBIC FOOT OF WATER BETWEEN 32° AND 212° F

Temperature Fahr.	Weight, lbs. per cubic foot	Temperature Fahr.	Weight, lbs. per cubic foot	Temperature Fahr.	Weight, lbs. per cubic foot
32°	62.42	123°	61.68	168°	60.81
35	62.42	124	61.67	169	60.79
40	62.42	125	61.65	170	60.77
45	62.42	126	61.63	171	60.75
50	62.41	127	61.61	172	60.73
52	62.40	128	61.60	173	60.70
54	62.40	129	61.58	174	60.68
56	62.39	130	61.56	175	60.66
58	62.38	131	61.54	176	60.64
60	62.37	132	61.52	177	60.62
62	62.36	133	61.51	178	60.59
64	62.35	134	61.49	179	60.57
66	62.34	135	61.47	180	60.55
68	62.33	136	61.45	181	60.53
70	62.31	137	61.43	182	60.50
72	62.30	138	61.41	183	60.48
74	62.28	139	61.39	184	60.46
76	62.27	140	61.37	185	60.44
78	62.25	141	61.36	186	60.41
80	62.23	142	61.34	187	60.39
82	62.21	143	61.32	188	60.37
84	62.19	144	61.30	189	60.34
86	62.17	145	61.28	190	60.32
88	62.15	146	61.26	191	60.29
90	62.13	147	61.24	192	60.27
92	62.11	148	61.22	193	60.25
94	62.09	149	61.20	194	60.22
96	62.07	150	61.18	195	60.20
98	62.05	151	61.16	196	60.17
100	62.02	152	61.14	197	60.15
102	62.00	153	61.12	198	60.12
104	61.97	154	61.10	199	60.10
106	61.95	155	61.08	200	60.07
108	61.92	156	61.06	201	60.05
110	61.89	157	61.04	202	60.02
112	61.86	158	61.02	203	60.00
113	61.84	159	61.00	204	59.97
114	61.83	160	60.98	205	59.95
115	61.82	161	60.96	206	59.92
116	61.80	162	60.94	207	59.89
117	61.78	163	60.92	208	59.87
118	61.77	164	60.90	209	59.84
119	61.75	165	60.87	210	59.82
120	61.74	166	60.85	211	59.79
121	61.72	167	60.83	212	59.76
122	61.70				

WEIGHT OF A CUBIC FOOT OF WATER AT HIGH TEMPERATURES

Temperature Fahr.	Weight of 1 cubic foot	Difference per 1 degree	Temperature Fahr.	Weight of 1 cubic foot	Difference per 1 degree	Temperature Fahr.	Weight of 1 cubic foot	Difference per 1 degree
220°	59.641	0.0253	290°	57.585	0.0318	350°	55.523	0.0360
230	59.372	0.0269	300	57.259	0.0326	360	55.158	0.0365
240	59.096	0.0276	310	56.925	0.0334	370	54.787	0.0371
250	58.812	0.0284	320	56.584	0.0341	380	54.411	0.0376
260	58.517	0.0295	330	56.236	0.0348	390	54.030	0.0381
270	*58.214*	*0.0303*	340	55.883	0.0353	400	53.635	0.0395
280	*57.903*	*0.0311*						


1140
1120
425
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415
395
1080
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375
355
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1020
120
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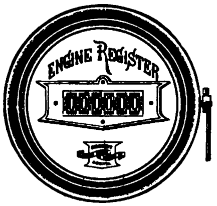
APPENDIX

COUNTERS, GAGES, AND OTHER ACCESSORY APPARATUS

THE CROSBY REVOLUTION COUNTER OR ENGINE REGISTER

Patented



Engine Revolution Counters, as commonly designed and constructed, depend upon an escapement for receiving the actuating force, and a star-toothed wheel for transmitting the movement to the figure-wheels. The escapement principle, while well adapted to a delicate machine actuated by a constant force, like a watch or clock, is ill suited to a counter for registering the stroke of a steam engine or other ponderous machine, where the actuating force may be out of all suitable proportion to the structural strength of the counter, and such as to destroy it, if from any cause, like the varying stroke of a pump, the pallet fails to properly engage with the tooth of the wheel. From the same causes such counters may also fail to register. These facts are well known to all mechanics who have had experience with escapement counters. To forcibly illustrate how poorly adapted to a counter the escapement principle is, it is only necessary to state, that in a watch, by the slight force of the actuating spring or weight, it simply *permits* a tooth to *escape*, while in a counter it is required to *move the whole mechanism*, and to do this intermittently and with the varying resistance of one or all of the figure-wheels.

In a Revolution Counter which shall be *reliable*, *durable*, and free from liability to serious injury, the actuating force must, through proper mechanism, be transmitted *directly* and with *certainty* to the figure-wheels, and this can best be done by means of a crank. It matters not in the Crosby Counter whether the movement of the crank is rotary or merely oscillatory, it will count just the same.

The Crosby Improved Revolution Counter, only, employs the crank principle, applied through other simple mechanical motions, so as to record with certainty the operations of any machine, and at the same time obviate all danger of injury to the counter itself or the machine to which it is attached.

This counter is adapted to either right or left hand rotary or reciprocating motions, and is capable of 500 revolutions per minute with safety to the machine and accuracy in the enumeration.

The shaft through which the actuating force is applied may extend from the counter either on the right-hand or left-hand side as desired.

It is made in the following sizes :

12 inch dial	8 wheels
12 “ “	7 “
12 “ “	6 “
10 “ “	8 “
10 “ “	7 “
10 “ “	6 “
8½ “ “	8 “
8½ “ “	7 “
8½ “ “	6 “
6¼ “ “	6 “
6 “ “	6 “

CROSBY SQUARE COUNTER

The actuating mechanism of this counter is positive and employs the principles just described, as used in the Crosby Revolution Counter. It is a strong and useful instrument, compact in form, durable and accurate. It may be provided with a re-setting device and also with a padlock if desired. When required for rotary motion, it should be stated whether it is to be used for right-hand or left-hand rotation.

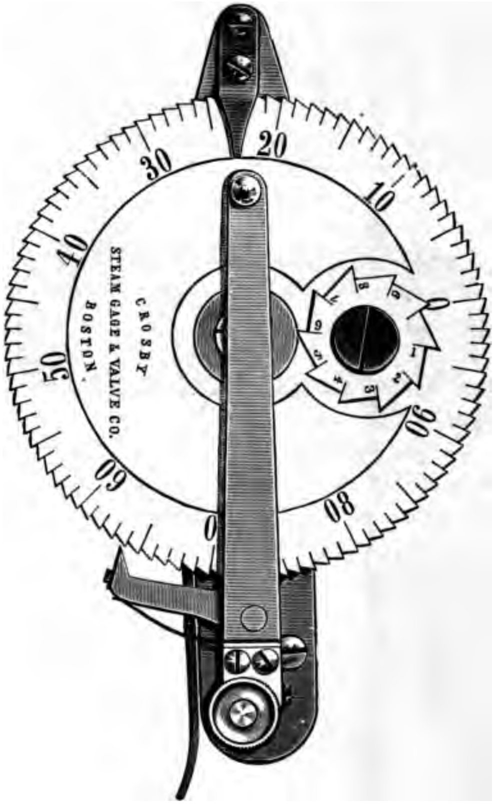
This counter is made in the following sizes :

4⅝ × 1½ in. dial	4 figures
5¼ × 1½ " "	5 "
6 × 1½ " "	6 "
7¼ × 2¼ " "	4 "
8¼ × 2¼ " "	5 "
9¼ × 2¼ " "	6 "
10⅜ × 2¼ " "	7 "

THE CROSBY LOCOMOTIVE COUNTER

For High Rotative Speeds

The cut on page 188 shows the Locomotive Counter. It is designed particularly for use on locomotives and high speed engines, and is a valuable auxiliary to the steam engine indicator. The arm which moves the ratchet is connected by a cord with some reciprocating part of the engine, or with the drum motion, so as to give it about 1½ inches swing back and forth during each revolution of the shaft. It is provided with a convenient starting and stopping device, so that it can be made to begin or stop counting at any instant.



CROSBY RECORDING COUNTER

Patented

This instrument furnishes a chart record of the revolutions or strokes of any engine, pump, or moving part. It is *designed with* remarkable genius for its special purpose, and it will record the highest speeds with mathematical accuracy. Every counter is tested to at least two thousand revolutions per minute, and will give positive results at much higher speeds without slip or error. Each regular chart affords a pen record up to fifty thousand consecutive strokes or revolutions, and the exact total number, or the elapsed count between any two noted periods, can be read with certainty. The highest working speeds found in mechanical operations are within the range of this device.

The Crosby Recording Counter is not a tachometer, but a chart-recording instrument, occupying a field by itself of peculiar importance. There is no other instrument like it, or that gives similar results. Its applications are varied and universal. It will be found especially valuable in making permanent records of the performance of machinery or engines either under special test or in daily service, and it is of the greatest usefulness and importance to every engineer, designer, and user of power. The chart is 8 inches in diameter and easily read; the mechanism is well constructed, durable, and accurate ; it cannot get out of order or adjustment. All like parts are interchangeable and suitably de-

signed to give proper wear and service. It is adapted for both revolutions and reciprocating motion without alteration. It is simple to attach and no skill is required to operate it.

A smaller recording counter capable of registering at the highest speeds upon a chart reading to 5,000 revolutions or strokes, and adapted to be attached to the Crosby Reducing Wheel, is described on page 73.

CROSBY PRESSURE AND VACUUM GAGES

Important

Accuracy is the essential feature in all gages, whether pressure or vacuum. The principle of construction of Crosby gages is correct and they embody important improvements in many essential details.

In the Crosby Improved Pressure Gage the tube springs are connected at each end with their respective parts by screw threads, without the use of any soldering material whatever, thus insuring tight joints under all conditions of heat and pressure.

The index mechanism and the dial are mounted upon an *extension of the* socket, thus rendering the entire operating parts of the gage independent of the case and free from any errors arising from its distortion or from external heat.

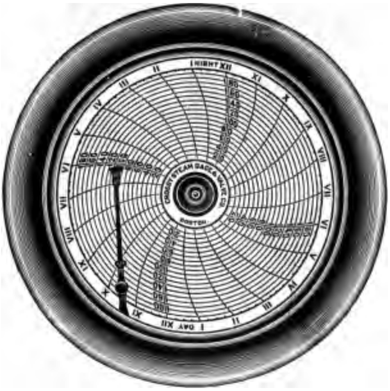
The method by which Crosby gages are tested and graduated will insure a truthful and reliable gage. Each one is tested under steam pressure and subjected to pressures accurately measured by standardized weights, and the gage is graduated to such absolute pressures and not by comparison only with another gage.

An equally accurate method is used in the testing of vacuum gages; each one is tried, marked, and adjusted by the direct readings of a mercury column, by means of an apparatus in which the successive stages of vacuum are actually produced.

Every gage used to indicate the pressure of steam should have a siphon or some other device which will furnish water to and completely fill the tube springs to keep them cool. Be sure that the connections between the gage and the siphon are perfectly tight.

CROSBY PRESSURE RECORDER

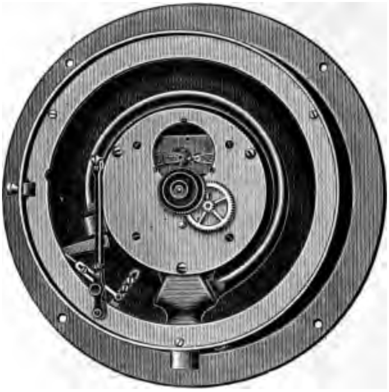
Patented



The Crosby Pressure Recorder records the pressure of any fluid during a certain period of time.

It is designed to supply the great and constantly increasing demand for a compact and reliable, yet not too costly, instrument for recording all the variations of pressure which take place in a steam boiler or other receptacle. It gives a graphic chart showing every such variation of pressure, its extent and duration, and the time when it occurs. The case is circular and is uniform in appearance with the other instruments usually set up in an engine room.

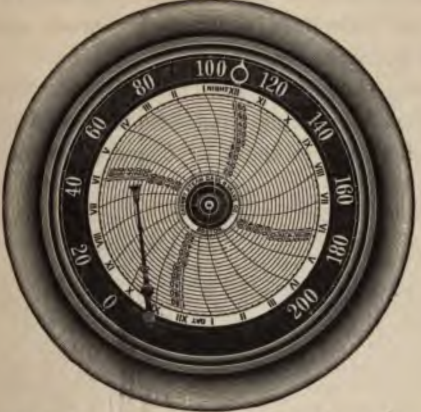
The socket by which it is attached to the boiler extends upward within the case, and supports the clock and other operating parts, thus producing a unity of action which gives a record true to the axis of the chart under all conditions. On the same post with the pen lever and below it, is a corresponding lever carrying a small table on which rests the pen point. Between the pen and table is placed the chart, and as both bear upon it there is insured a continuous contact and in operation a continuous line.



The pen is charged with a supply of red ink, and is easily and delicately adjusted to the surface of the chart by an ingenious device at its base, giving records of unequaled *legibility* and accuracy.

CROSBY PRESSURE RECORDER AND GAGE

Patented



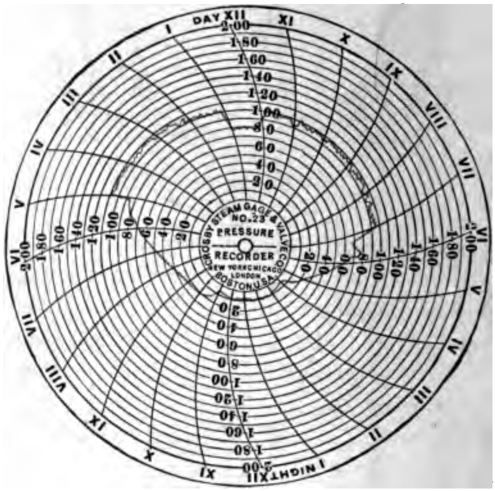
The Crosby Pressure Recorder and Gage, in addition to recording the pressure of any fluid during a certain period of time, has an index hand and an outside circle of figures to show the actual pressure recorded on the chart at the moment of observation; in this respect it operates like an ordinary steam gage.

The chart rotates once in 24 hours, and by the employment of a special clock movement, the rotation of the chart may be made to conform to any period of time from one hour to one week, thus adapting the instrument to almost any conditions to be found, in which a record of pressures is desirable or necessary; and the range of pressures which may be recorded is practically unlimited.

The reading of the chart, as shown above, is 110 pounds pressure at 6.30 o'clock A.M. Supposing the instrument to be properly connected to a steam boiler, or other receptacle, then during the next 24 hours, a red line is traced by the pen completely around the dial, showing by its deviations from a true circle, the variations of pressure which take place during the whole of that time.

Special Crosby Recording Gages are made to give continuous chart records of the working pressures in air-brake reservoirs and train lines, and in the cylinders and tanks of pumps and hydraulic presses.

The Crosby Gas, Mine, and Draft Recorder is an instrument designed for making a continuous record of the pressure of fluids, either above or below the atmosphere, as ordinarily measured in inches of water. It is useful for determining and recording the drafts of chimneys, or the pressure of air in the ash pit of a steam boiler, in mines when forced therein, or in buildings when introduced for heating and ventilation; it will show the pressure of gas for illuminating or other purposes at the works or at the place of consumption, or of any fluid where the pressure is sought and its record desired. This instrument is similar in the character of its work or operations to the Crosby Recording Gage, but is adapted to conditions requiring one which is *more sensitive* and delicate.



Reproduction of Actual Chart

Scale, one-twelfth size

Steam Pipe Thermometer

CROSBY
FAH'T. TEMPERAT.
PRESSURE LBS. PR. SQ. IN.
270
250
230
210
190
170
150
130
30
20
10
0
CROSBY STEAM GAGE & VALVE CO. BOSTON

Hot Water Thermometer

Hot Well Thermometer

THE CROSBY INDICATOR

It may be of interest to those who have read this book to know that whatever of merit in this instrument has been described and illustrated in these pages has been recognized and acknowledged as follows:

At the Paris Exposition of 1889, where it received the highest award, a gold medal.

At the World's Columbian Exposition, Chicago, in 1893, where it received the highest award, a medal and diploma.

At the Cotton States and International Exposition at Atlanta, in 1895, where it received the highest award.

At the Russian Exposition held at Nijni Novgorod, in 1896, where it received the highest award, a gold medal and diploma of honor.

At the Louisiana Purchase Exposition, held at St. Louis in 1904, where it received the grand prize.

The Crosby Indicator is approved and adopted by the United States Government. It is the standard in nearly all the great electric light and power stations of the United States. It has been approved and adopted by the principal navies, the government shipyards, and the most eminent technical schools of the world.

Full particulars for the proper care and handling of the Crosby Indicator accompany each instrument.

CROSBY STEAM GAGE AND VALVE COMPANY

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